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# LIGHTWEIGHT SIDEWALLS FOR AIRCRAFT INTERIOR NOISE CONTROL

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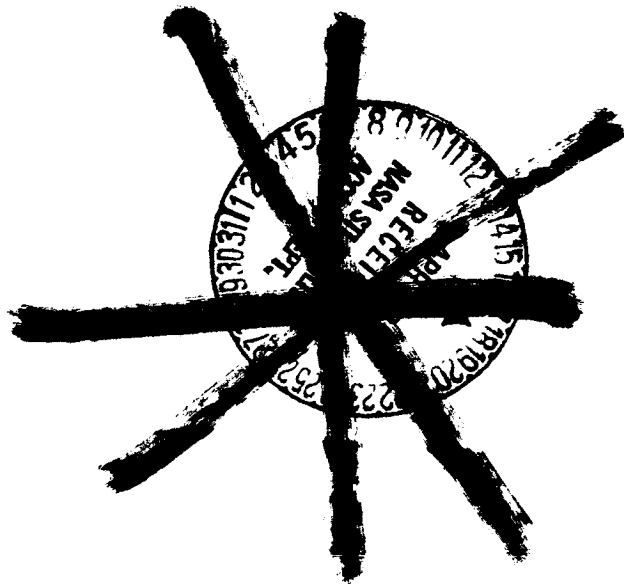
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D. N. May, K. J. Plotkin, R. G. Selden and B. H. Sharp

McDonnell Douglas Corporation  
Douglas Aircraft Company  
Long Beach, CA 90846

CONTRACT NAS1-17263  
FEBRUARY 1985



**NASA**

National Aeronautics and  
Space Administration

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## FOREWORD

This study was performed for NASA Langley Research Center by Douglas Aircraft Company (under Contract NAS1-17263) and by Wyle Laboratories as subcontractor under Agreement No. AS-21958-C. Douglas was responsible for the definition of the noise control requirement, sidewall panel detail design and construction, acoustical testing and data reduction, and the nonacoustical assessment. Wyle was responsible for sidewall panel concept identification, analysis and conceptual design, and for acoustic results interpretation. The NASA Langley technical monitor was J. S. Mixson.

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## 1.0 INTRODUCTION

The purpose of this study was to identify and test concepts for achieving high values of noise reduction in aircraft sidewalls without reliance on addition of mass or other conventional techniques, in a form that may ultimately be suitable for application to advanced turboprop commercial transports. The propellers of these aircraft are expected to generate low frequency sound levels on the fuselage far higher than generally found with turbofan aircraft. Their sound field presents a difficult challenge to achieving comfortable cabin noise levels without significant sidewall weight or wall depth penalties as compared with current design practices.

The study had, as a focus, commercial aircraft of conventional primary structure (aluminum skin-stringer construction) with 3 m (10 ft) to 4 m (13 ft) fuselage diameters, i.e., narrow body aircraft with 4- to 6-abreast coach-class passenger seating. Therefore only sidewalls with depths of 7.5 cm (3.0 in) or less were considered. Since, within a one man-year level of effort, the goal was to examine a wide range of advanced sidewall concepts rather than fully develop a limited number, flat rather than curved test panels and simple non-metallic constructions without windows were often employed in the testing.

The study progressed through several stages:

### (1) Transmission Loss Characteristics of Structures (Section 2)

In this stage a basic study was performed of the characteristics of sound transmission through structures. All structures were basically either single panels, multi-layer panels, multiple panels, or a combination of the three, although other sound-attenuating techniques were sometimes superimposed. Each of these basic structures, and their combinations, were analyzed and the effect of all parameters evaluated. Simple methods of analysis were used, as opposed to the more complex modal methods, so as to quickly assess the benefits of different configurations.

To guide the direction of the analyses and selection of suitable techniques, the exterior sound spectrum (incident sound) was assumed to be as shown in Figure 1. The spectrum was derived from propeller manufacturer design curves, and is characterized by a series of harmonically related peaks, with a fundamental of 160 Hz. This corresponds to a maximum overall sound pressure level (incident plus reflected waves, assuming a 5 dB increment for the reflection) of 145 dB or an A-weighted level of 135.5 dB. This sound field was predicted to occur, at Mach 0.8 cruise, on the fuselage of an advanced turboprop aircraft having a 10-bladed single-rotating propeller of 4.2 m (13.9 ft) diameter with a tip speed of 213 m/s (700 ft/s), and with a propeller tip to fuselage spacing of 0.8 propeller diameters. The study had as focus an interior noise goal of 80 dB(A). The required A-weighted noise reduction based on the incident sound wave was therefore about 50 dB, implying transmission loss values in the range 40 to 60 dB with emphasis on frequencies below 500 Hz.

## (2) Test Panel Design and Construction (Section 3)

In this stage of the study 1.2 m by 1.8 m (4 ft by 6 ft) test panels were designed and built to demonstrate the high transmission loss principles analyzed and found promising at the previous stage. Also provided were reference or comparison test panels from current aircraft practice.

## (3) Acoustical Testing (Section 4)

A simple noise reduction test technique was then used to measure the acoustical performance of the test panels.

## (4) Acoustical Interpretation (Section 5)

The noise reduction performances of the test panels were compared with each other and with prediction, and conclusions were drawn as to the functioning of the designed-for noise control features.

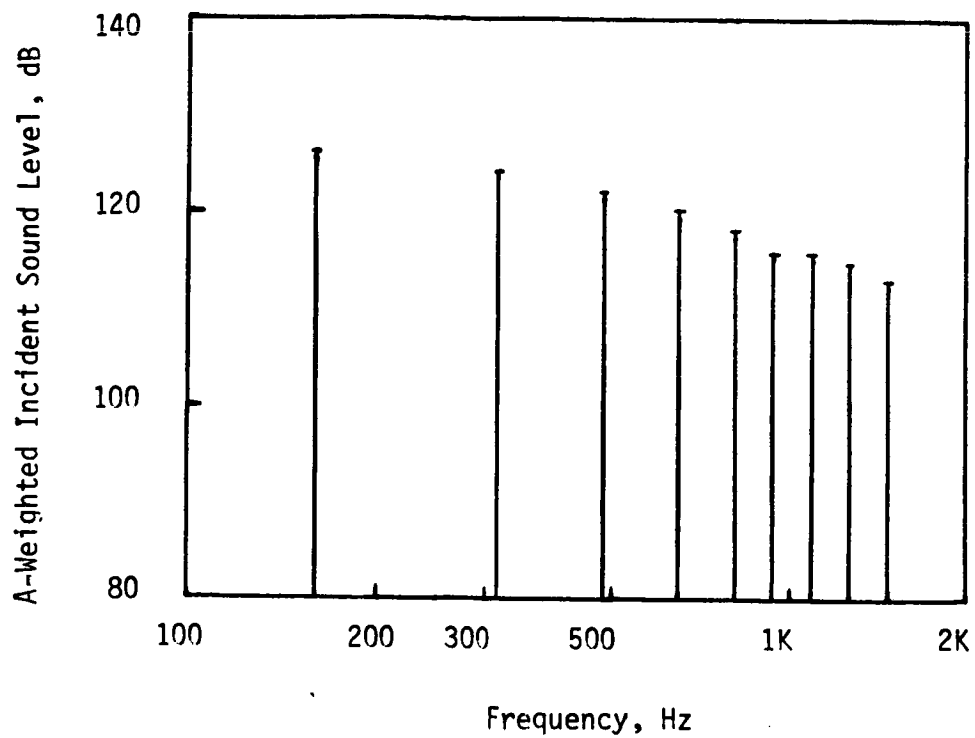


Figure 1. Fuselage Incident Turboprop Spectrum

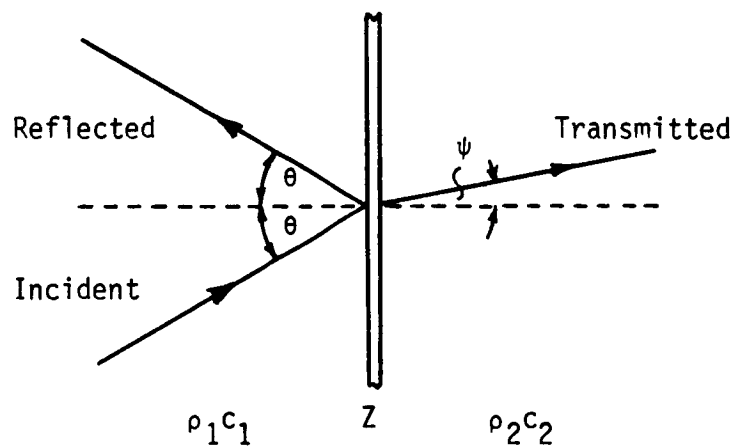


Figure 2. Sound Transmission through a Panel

## (5) Aircraft-Application Designs (Section 6)

The test panels were examined to establish aircraft-application designs which most closely met aircraft nonacoustical requirements; then these designs were adjusted to just meet the study aircraft's acoustical requirement and thus allow the designs to be compared on the basis of weight.

### 2.0 TRANSMISSION LOSS CHARACTERISTICS OF STRUCTURES

The mechanisms of sound transmission through materials and the prediction of the transmission loss of different types of structures are briefly analyzed in this section, to provide a basis for designing high transmission loss test panels incorporating concepts other than mass. A summary of the conclusions of this analysis is given in Section 2.6.

#### 2.1 Single Panels

The simplest type of structure to consider is a single panel whose thickness is small compared to the associated airborne and structureborne wavelengths. If the panel is infinite in size, i.e., the dimensions are much greater than the wavelength of bending waves, it can be shown by classical methods<sup>1</sup> that the transmission coefficient  $\tau_\theta$ , defined as the ratio of transmitted to incident sound power, for sound waves incident at a single angle  $\theta$ , as shown in Figure 2, is given by the expression:

$$\tau_\theta^{-1} = \frac{|\rho_1 c_1 \cos \psi + \rho_2 c_2 \cos \theta + Z \cos \theta \cos \psi|^2}{4 \rho_1 c_1 \rho_2 c_2 \cos \theta \cos \psi} \quad (1)$$

where  $\rho_1 c_1$ ,  $\rho_2 c_2$  are the characteristic impedances of the media on either side of the panel ( $\rho$  = density,  $c$  = speed of sound),  $\theta$  is the angle of incidence,  $\psi$  is the angle of radiation,  $\theta$  and  $\psi$  are related by Snell's Law, and  $Z$  is the specific impedance of the panel given by the expression:

$$Z = i\omega m - \frac{i\omega^3 B}{c^4} \sin^4 \theta \quad (2)$$

where  $m$  is the mass of the panel per unit area,  $\omega$  is the radial frequency ( $= 2\pi f$ ), and  $B$  is the bending stiffness of the panel material.\* The transmission loss,  $TL$ , of a panel is given by the expression  $TL = 10 \log (1/\tau_\theta)$ .

At low frequencies, the impedance  $Z$  is dominated by the inertial impedance  $\omega m$ , giving the familiar mass-law where the transmission loss increases at a rate of 6 dB per octave. At high frequencies, the bending stiffness term dominates the impedance term. At some intermediate frequency, the mass and bending stiffness terms are equal in magnitude and opposite in sign, so that in the absence of damping, the panel impedance  $Z$  is zero. The frequency at which this occurs is termed the coincidence frequency, given by the expression:

$$f_{\text{coincidence}} = (c^2/2\pi \sin^2 \theta) (m/B)^{1/2} \quad (3)$$

In practice, the damping is never zero, but the frequency response for the impedance  $Z$ , and hence the transmission loss, exhibits a noticeable dip at this frequency. For any given panel, the frequency at which coincidence occurs depends on the angle of incidence,  $\theta$ , of the sound. The lowest coincidence frequency occurs at grazing incidence ( $\theta = \pi/2$ ), and is known as the critical frequency,  $f_c$ , given by the expression:

$$f_c = (c^2/2\pi) (m/B)^{1/2} \quad (4)$$

For other angles of incidence, coincidence occurs at a frequency equal to  $f_c/\sin^2 \theta$ .

If, as usual, the impedance of the panel is much greater than the characteristic impedance of the two media, the transmission loss  $TL_M$  of a single panel for sound incident at an angle  $\theta$ , at frequencies less than the coincidence frequency, is given by the expression:

---

\*The convention used throughout this report is that  $i = \sqrt{-1}$ , and  $+i\omega m$  represents a mass reactance.



$$TL_M = 20 \log (mf) - 10 \log (\rho_1 c_1 \rho_2 c_2) + 10 \log (\cos \theta \cos \psi) + 10, \text{dB} \quad (5)$$

The dependence of  $TL_M$  on  $\rho_1 c_1$  and  $\rho_2 c_2$  shows that the transmission loss of an aircraft structure will vary with altitude, increasing as  $\rho c$  decreases with increasing altitude. For example, the difference in  $TL_M$  for a given panel at sea level and at 9,100 m (30,000 ft) is 5.4 dB (assuming a cabin pressurization equivalent to 1,500 m (5,000 ft)).

At frequencies greater than the coincidence frequency, the rate of increase of transmission loss with frequency ranges from 6 dB to 18 dB per octave depending on the angle of incidence. For random sound incidence, as would be approximated in a reverberation chamber, the coincidence effect is exhibited as a dip in the transmission loss curve at the critical frequency, and the rate of increase at higher frequencies is approximately 9 dB per octave.

For finite-sized panels, it is necessary to include an additional term in the expression for impedance to account for the stiffness of the panel. This term is important only at frequencies below the region where panel resonances occur. There are of course, a large number of panel resonances, but except for the case where the panel damping is low, the most important as far as sound transmission is concerned is the fundamental resonance,  $f_{11}$ . The addition of a stiffness factor to the panel impedance results in an additional term  $20 \log (f_{11}/f)$  in the expression for transmission loss in Equation (5).

For a simply-supported panel, the fundamental resonant frequency is given by the expression:<sup>2</sup>

$$f_{11} = \frac{\pi}{2} \sqrt{\frac{B}{m}} \left( \frac{1}{a^2} + \frac{1}{b^2} \right), \text{Hz} \quad (6)$$

where  $a$  and  $b$  are the dimensions of the panel. Since both Equations (4) and (6) contain the term  $\sqrt{B/m}$ , it is easy to show that

$$f_{11} \cdot f_c = \frac{c^2}{4} \left( \frac{1}{a^2} + \frac{1}{b^2} \right) \quad (7)$$

Thus the product of the fundamental panel resonance (simply-supported) and the critical frequency is independent of the properties of the panel and is a function only of the panel dimensions and the speed of sound.

The general form of the transmission loss curve for a finite-sized panel as a function of frequency is shown in Figure 3 for random incidence sound. For an aircraft fuselage skin of 1.6 mm (0.063 in) aluminum on a typical frame/stringer configuration, the critical frequency is on the order of 8000 Hz, and the fundamental panel resonance occurs at about 125 Hz. Thus, over much of the frequency range of interest in this study, a typical baseline fuselage structure is in the mass-law region where increases in transmission loss have conventionally only been achieved by increasing the mass. The following sections of this chapter explore alternative structural configurations which are not in the mass law region over the frequency range of interest, and which may achieve higher values of transmission loss without significant increases in mass.

## 2.2 Thick Single and Multi-Layer Panels

The expressions for transmission loss given in the previous section are correct only in the frequency range where the thickness of the panel is small compared to the wavelength of bending waves. According to Cremer<sup>3</sup>, the "thin" panel theory is correct provided the bending wavelength is at least six times greater than the panel thickness. At higher frequencies, the equations of the previous section must contain a correction for the influence of shearing motion in the panel. The type of wave motion, either bending or shearing, that predominates at any given frequency is the one that provides the lowest impedance to the incoming sound field.

Sharp<sup>4</sup> has shown that the general impedance,  $Z$ , of a single panel can be expressed as:

$$Z = i\omega m + Z_B Z_S / (Z_B + Z_S) \quad (8)$$

where  $Z_B$  is the impedance for bending waves as given by the second term in Equation (2), and  $Z_S$  is the impedance for shear waves given by the following expression:

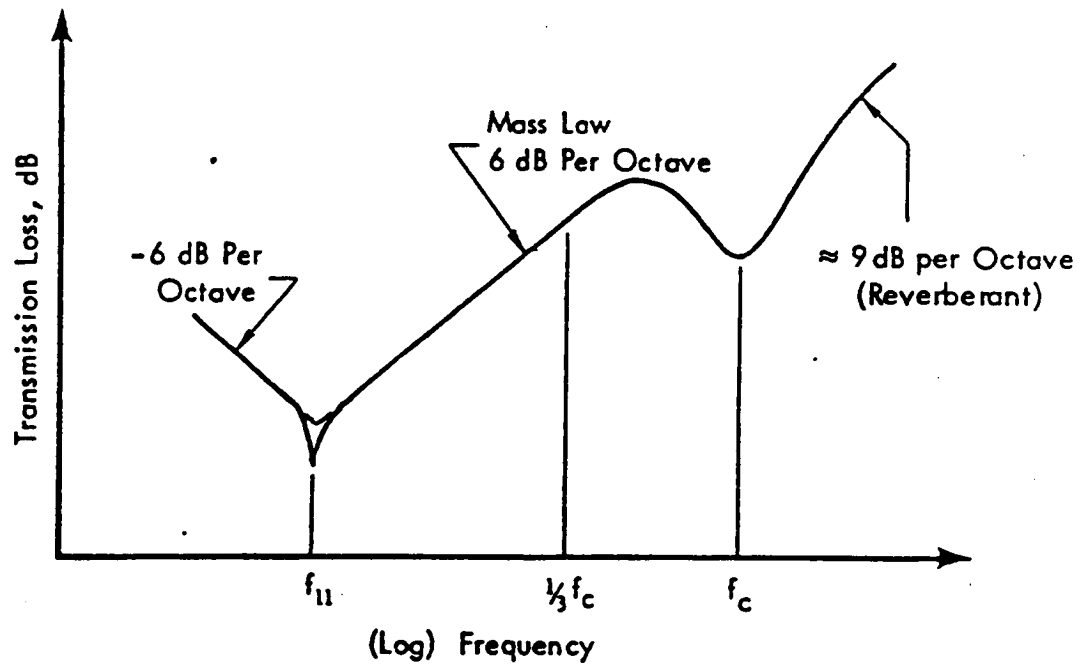


Figure 3. The General Form of the Transmission Loss as a Function of Frequency for a Finite Size, Thin Single Panel

$$Z_S = -i \frac{\mu h \omega}{c^2} \sin^2 \theta \quad (9)$$

where  $\mu$  is the shear modulus of the panel material, and  $h$  is the thickness of the panel. This modification of the thin single-panel theory can also be used to predict the performance of sandwich structures where shearing occurs only in the core material.

### 2.2.1 High Stiffness Multi-Layer Panels

Examination of Figure 3 shows that the transmission loss of a single panel exceeds that predicted by the mass law at some frequencies greater than the critical frequency  $f_c$ . Thus, if the critical frequency of a panel was very low, say 100 Hz, then in theory the mass law could be exceeded over a considerable frequency range. The feasibility of designing such a panel is discussed below, first by examining the panel stiffness requirements, and second by determining the necessary core parameters.

For a panel to exhibit a critical frequency of 100 Hz, Equation (4) shows that the ratio of bending stiffness to mass  $B/M$  must equal  $3.4 \times 10^4 \text{ m}^4/\text{sec}^2$ . Such a high ratio can only be practically achieved by a multi-layered (sandwich) panel with two face plates and a rigid core (such as honeycomb). If the total surface mass of the panel is  $7.4 \text{ kg/m}^2$  ( $1.5 \text{ lb/ft}^2$ ), and the mass of the core is negligible, then the surface mass of each face is  $3.7 \text{ kg/m}^2$  - equivalent to a 1.4 mm (0.054 in) aluminum panel. The bending stiffness  $B$  of a sandwich panel is given by the expression:

$$B = E \left[ \frac{h^3}{6} + \frac{h}{2} (h + d)^2 \right] \quad (10)$$

where  $E$  is the Young's modulus,  $h$  is the thickness of the aluminum face plates (assumed equal), and  $d$  is the thickness of the core. If the core thickness is 2.54 cm (1 in), then  $B/M$  is equal to  $5 \times 10^3 \text{ m}^4/\text{sec}^2$ , or a factor of about 7 less than required to achieve the required critical frequency. To obtain a critical frequency of 100 Hz with 1.4 mm aluminum face plates requires that the core be about 7 cm (2.75 in) thick. Alternatively, the mass of the panel, i.e., the face plates, could be decreased by a factor of 7. However,

with this arrangement, the absolute value of the transmission loss would be very low.

The above discussion assumes that the core exhibits bending motion only, and that the shearing impedance of the composite structure is extremely high. The required shear modulus for the core can be determined by examination of Equation (8). For a stiff panel without shear, the condition  $Z_S > Z_B$ , must be met over the required frequency range. The impedance  $Z$  of the panel with the core exhibiting a shearing wave motion is given by the expression:

$$Z = 2Z_{B1} + Z_S \quad (11)$$

where  $Z_{B1}$  is the bending impedance of each of the face plates, and  $Z_S$  is the shearing impedance of the core. Using values for these impedances given in Equations (2) and (9), respectively, the inequality  $Z_S > Z_B$  can be restated as follows:

$$2 Z_{B1} + Z_S > Z_B$$

or

$$\frac{2 \omega^3 B_1}{c^4} + \frac{\mu d \omega}{c^2} > \frac{\omega^3 B}{c^4} \quad \text{for } \theta = \frac{\pi}{2} \quad (12)$$

where  $B_1$  is the bending stiffness of the face plates. Incorporating the expressions for the critical frequencies of the face plates ( $f_{c1}$ ) and for the overall panel ( $f_c$ ):

$$f_c = \frac{c^2}{2\pi} \left( \frac{2m}{B} \right)^{1/2}$$

$$f_{c1} = \frac{c^2}{2\pi} \left( \frac{m}{B_1} \right)^{1/2}$$

Equation (12) can be rewritten as follows:

$$\frac{\mu d}{2mc^2} > \left(\frac{f}{f_c}\right)^2 - \left(\frac{f}{f_{c1}}\right)^2$$

Since  $f_{c1} \gg f_c$ , the condition for  $Z_S > Z_B$  is:

$$\frac{d}{2mc^2} > \left(\frac{f}{f_c}\right)^2 \quad (13)$$

In the present case,  $f_c$  is required to be 100 Hz. If shearing of the core is to be minimized at frequencies below 1500 Hz, then  $(f/f_c)^2 = 225$ . For a composite panel 2.54 mm (1 in) thick with a surface weight of 7.4 kg/m<sup>2</sup> (1.5 lb/ft<sup>2</sup>),

$$\mu = 7.6 \times 10^9, \text{ N/m}^2$$

This value of the shear modulus is higher than generally available from honeycomb structures.

In summary, it appears that stiff honeycomb sandwich panels applicable to aircraft fuselages cannot be designed to have a critical frequency on the order of 100 Hz. For a reasonable maximum thickness of 1.27 cm (0.5 in), the critical frequency is on the order of 500 Hz. It can be shown<sup>1</sup> that the transmission loss of a stiff panel exceeds that given by the mass law only at frequencies greater than

$$f = \frac{0.45 f_c}{\eta}$$

where  $\eta$  is the damping factor of the composite panel. Thus, if  $\eta = 0.1$ , a value requiring a considerable increase in damping, the mass law would be exceeded only at frequencies greater than 2250 Hz. Furthermore, this assumes that shearing of the core does not occur - not a good assumption according to the results described above. Accordingly, this type of structure cannot be designed to provide values of transmission loss greater than those given by the mass law at low and medium frequencies under the constraints imposed for aircraft fuselage application.

### 2.2.2 Variable Stiffness Multi-Layer Panels

The requirements for high static stiffness to withstand loads and high critical frequency for high sound transmission loss are incompatible in a single panel - see Equation (4). It is possible, however, to achieve these two goals in a multi-layer panel with careful design of the component parameters, as originally shown by Kurze.<sup>5</sup> The application of this type of panel to aircraft fuselage use is examined below by determining the core parameters necessary to meet the stiffness requirements and to minimize panel resonances.

Examination of Equation (8) shows that when shearing motion of the panel predominates, i.e.,  $Z_S < Z_B$ , the panel impedance is given by the expression:

$$Z = 2i\omega m + 2 Z_{B1} + Z_S \quad (14)$$

using Equation (11). Inserting the expressions for  $Z_{B1}$  and  $Z_S$  as before, and letting  $Z = 0$ , the critical frequency  $f_s$  of the composite panel in shearing motion is given by:

$$f_s = f_{c1} \left(1 - \frac{\mu d}{2mc^2}\right)^{1/2} \quad (15)$$

Using the same material constants as in the previous example, i.e., 1.4 mm aluminum face plates with a spacing of 2.54 cm, together with a core having a shear modulus of  $5 \times 10^6$  N/m<sup>2</sup> (an average value for a rigid polyurethane foam), gives the value of  $f_s$  as:

$$f_s = 0.92 f_{c1}$$

Therefore, if the panel is designed to allow shearing of the core to occur at a frequency less than the critical frequency  $f_c$  of the composite panel, the coincidence effect will be avoided and the effective critical frequency will be similar to that of the face plates. Under this condition, the panel will obey the mass-law over a wide frequency range. If bending of the panel

predominates at  $f = f_c$ , then the coincidence effect will occur.

For a multi-layer panel, the requirements for panel stiffness can be stated as follows:

(1)  $Z_S > Z_B$  for  $f \ll f_c$  to provide static rigidity

(2)  $Z_S < Z_B$  for  $f = f_c$  to allow core shearing

where  $f_c$  is the critical frequency of the composite panel in the absence of shearing.

Proceeding as before in the previous section, the first condition (1) results in the following inequality:

$$\frac{\mu d}{2mc^2} > \left( \frac{f}{f_c} \right)^2 \quad (16)$$

In this case, however,  $f \ll f_c$  because the shearing stiffness needs to be high at zero frequency to provide a panel with a high static rigidity. Approximating this condition by assuming that  $f = 0.1 f_c$ , Equation (16) can be reduced to

$$\frac{\mu d}{2mc^2} > 0.01 \quad (17)$$

The second condition (2) expresses the requirement that shearing motion of the panel shall be more strongly excited than bending wave motion at frequencies less than  $f_c$ . This requirement is satisfied provided that

$$\frac{\mu d}{2mc^2} < 1 - \left( \frac{f_c}{f_{c1}} \right)^2$$



where  $f_{c1}$  is the critical frequency of the face plates. Since  $f_{c1} \gg f_c$ , this inequality can be expressed approximately as follows:

$$\frac{\mu d}{2mc^2} < 1 \quad (18)$$

Combining the two conditions (1) and (2) gives the requirement that

$$0.01 < \frac{\mu d}{2mc^2} < 1$$

$$2.5 \times 10^3/d < \frac{\mu}{m} < 2.5 \times 10^5/d, \text{ m/s}^2 \quad (19)$$

A suitable value for  $\mu/m$  would therefore appear to be  $2.5 \times 10^4/d \text{ m/sec}^2$ , the mid-point of the inequality.

In addition to the requirements for bending stiffness, it is also necessary to ensure that the fundamental "mass-spring-mass" resonance,  $f_0$ , lies outside the frequency range of interest. The expression for  $f_0$  is:

$$f_0 = \frac{1}{2\pi} \left( \frac{2K}{md} \right)^{1/2} \quad (20)$$

where  $K$  is the compression modulus of the core material. For a typical case where Poisson's Ratio is 0.3, the core stiffness  $K \approx 4\mu$  (assuming that the core is isotropic)<sup>3</sup>. Inserting this value in Equation (20), and requiring that the resonance  $f_0$  be at a frequency greater than 3000 Hz, leads to the requirement:

$$\frac{\mu}{m} > 4.5 \times 10^7 \times d, \text{ m/s}^2 \quad (21)$$

Figure 4 shows curves of the quantity  $\mu/m$  plotted against the core thickness  $d$ , representing Equations (19) and (21), with arrows indicating the area of the graph in which values of  $\mu/m$  meet the criteria. The mid-point of the inequality of Equation (19) is also shown, representing the most suitable value of  $\mu/m$ . Thus for a core thickness in the range 1 to 2.5 cm, the most suitable value for  $\mu/m$  is about  $2 \times 10^6 \text{ m/s}^2$ . With a face plate surface

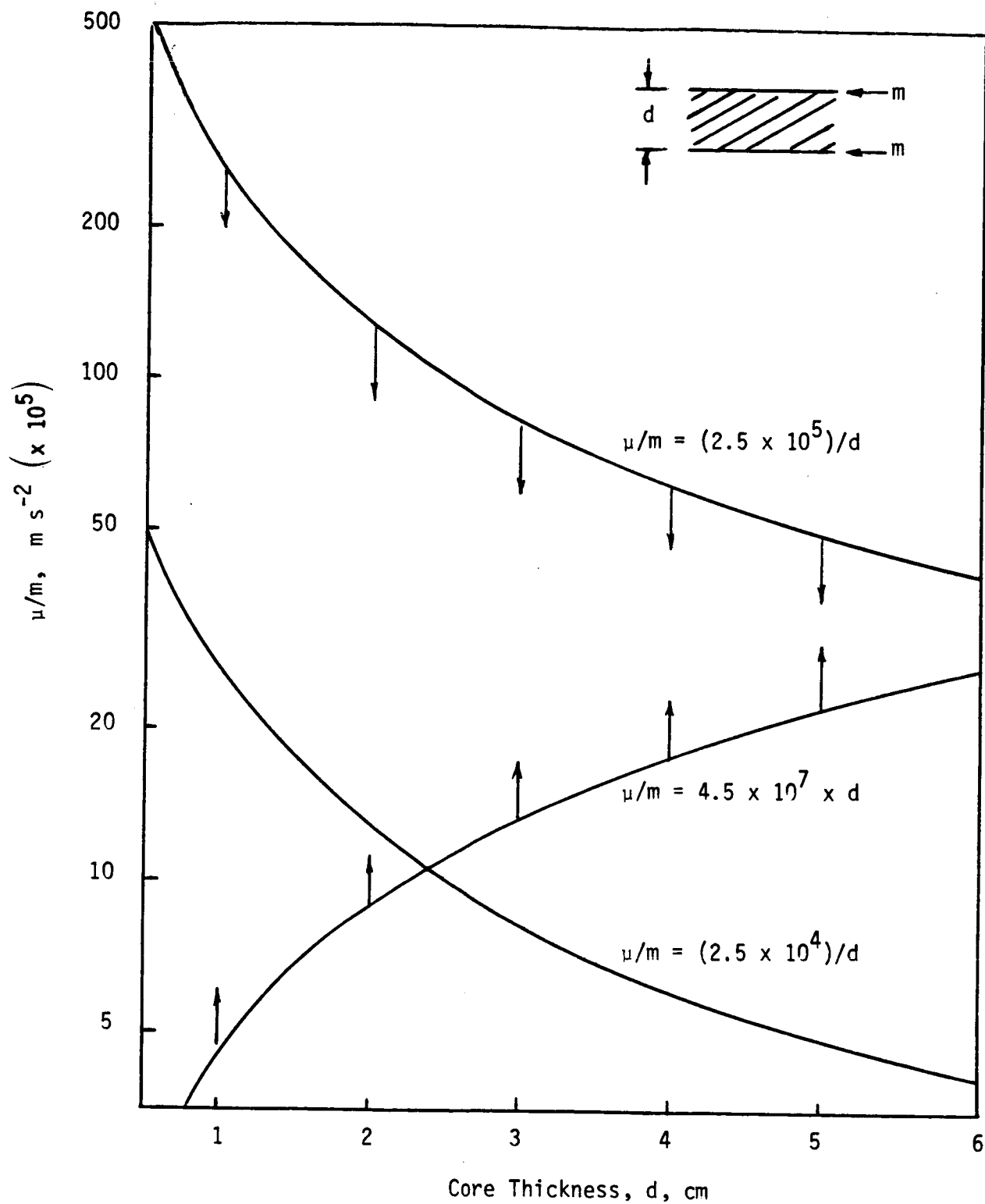


Figure 4. Design Graph for Selection of Material Parameters for a Variable Stiffness Sandwich Panel.

weight of  $3.7 \text{ kg/m}^2$ , this leads to the requirement that  $\mu = 7.4 \times 10^6 \text{ N/m}^2$ .

The analysis performed above indicates that a multi-layer panel can be designed to provide a transmission loss essentially equal to that predicted by the mass law, and yet exhibit high static stiffness suitable for withstanding loads.

The transmission loss of the multi-layer panel (total surface weight  $7.4 \text{ kg/m}^2$ ) is in accordance with the mass law, rising from approximately 23 dB at 250 Hz to 35 dB at 1000 Hz. To obtain higher values, it is necessary to increase the mass of the face plates. According to the mass law, doubling the thickness of the face plates will result in a 6 dB increase in transmission loss. To maintain the optimum core condition, the value of shearing modulus  $\mu$  must then also be increased.

The core material does not have to be isotropic - in fact, in the search for suitable materials it may be useful to select a material with a high compressional stiffness perpendicular to the panel faces (to maintain a high value for  $f_0$  - see Equation (20)), and a lower shear stiffness parallel to the faces, provided that this arrangement is consistent with the required static stiffness.

## 2.3 Multiple Panel Constructions

### 2.3.1 Double Panels

One method of obtaining higher values of transmission loss than available from a single panel is by the introduction of an additional panel with an intervening cavity. The frequency characteristics of such a construction are naturally more complex than for a single panel because the transmission loss is dependent on a greater number of parameters. The general form of the transmission loss curve for a double panel with absorption in the cavity is shown by the solid line in Figure 5. At low frequencies, the construction obeys the "mass law" of Equation (5) where the mass in this case is the combined mass of the two panels. The fundamental panel-cavity resonance occurs at a frequency  $f_0$  given by the expression:

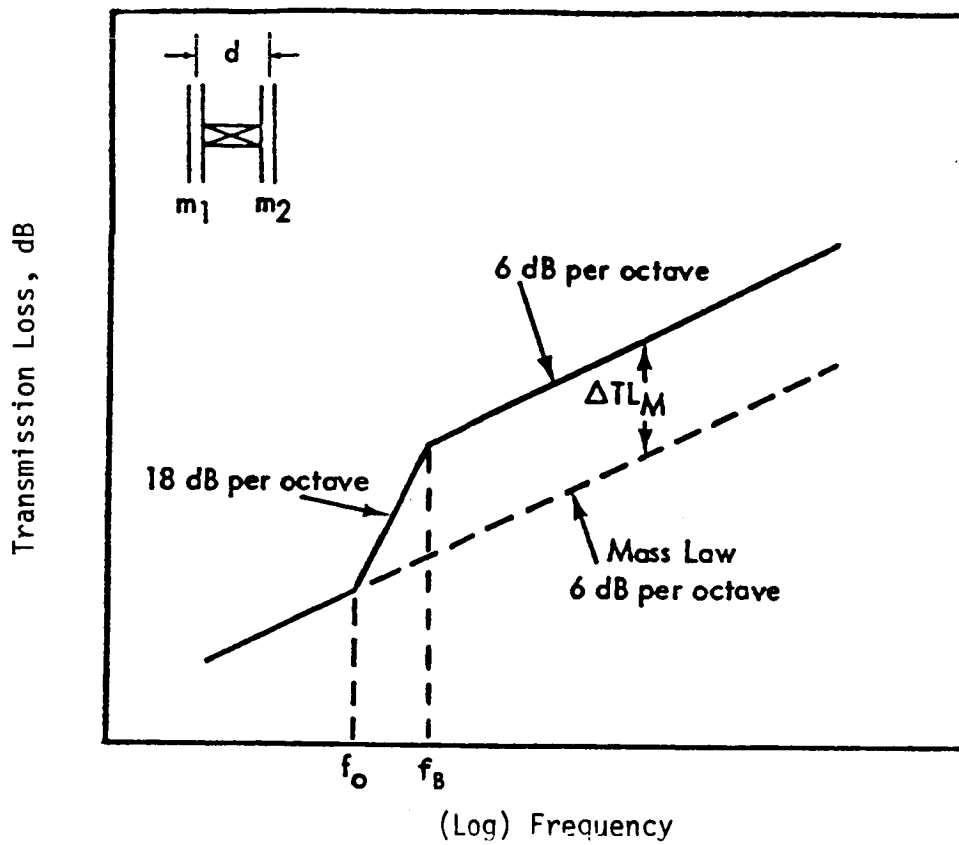


Figure 5. General Form for the Transmission Loss of a Double Panel with Sound Bridges

$$f_0 = \frac{1}{2\pi \cos \phi} \left[ \frac{\rho_2 c_2^2 (m_1 + m_2)}{m_1 m_2 d} \right]^{1/2}, \text{ Hz} \quad (22)$$

where  $m_1$  and  $m_2$  are the masses of the two panels per unit area,  $d$  is the panel separation,  $\rho_2$  and  $c_2$  are the air density and speed of sound in the cavity, and  $\phi$  is the angle of incidence in the cavity. This resonance causes a dip in the transmission loss curve, the magnitude of which is greatly diminished if there is acoustic absorption in the cavity.

At frequencies greater than  $f_0$ , the transmission loss of the double panel increases at a rate of 18 dB per octave, and would continue at this rate were it not for the effect of mechanical connections, or "bridges", between the panels that transfer vibrational energy from one panel to the other. The effect of bridges, which are required to provide rigidity, is to reduce the transmission loss at a rate of 12 dB per octave, with the result that, above the bridging frequency  $f_b$ , the net transmission loss increases at a rate of 6 dB per octave<sup>4</sup>, as shown in Figure 5. The transmission loss curve is parallel to an extension of the mass-law curve, but is higher by the amount  $\Delta TL_M$ . The value of  $\Delta TL_M$  dependent on the type and number of connections, and the critical frequencies of the two connected panels. For a simple case where two identical panels with critical frequency  $f_c$  are connected by frames with a regular spacing  $b$  meters, the value of  $\Delta TL_M$  is given by the expression:<sup>4</sup>

$$\Delta TL_M = 10 \log (bf_c) - 25, \text{ dB} \quad (23)$$

### 2.3.2 Effect of Major Parameters

An examination of Figure 5 shows that the transmission loss of a double panel exceeds that of a single panel, of the same total mass, only at frequencies greater than  $f_0$ . Therefore the major goal in designing a double panel for high transmission loss is twofold, namely:

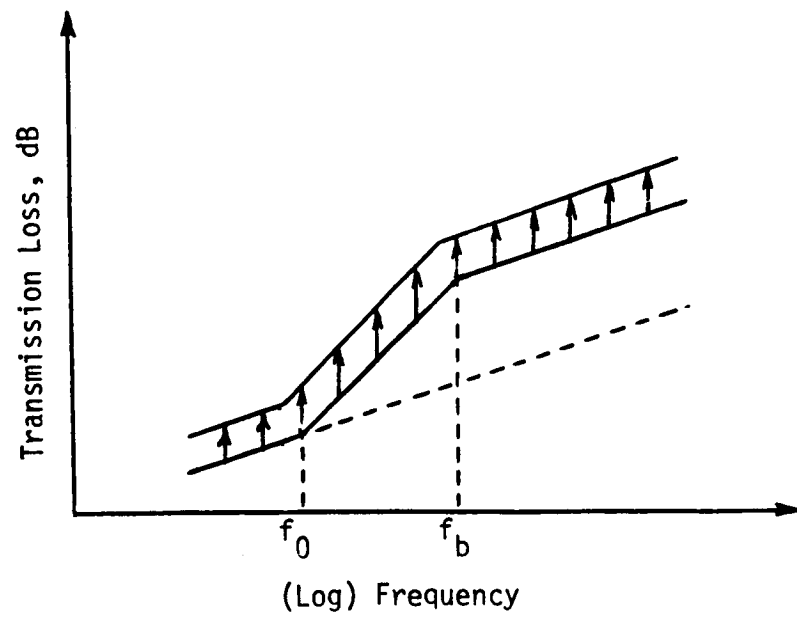
1. Minimize  $f_0$  so that it is below the frequency range of interest, and
2. Maximize  $\Delta TL_M$  over the entire frequency range.

Methods of achieving the first goal are apparent from inspection of Equation (22). First, it can be shown that for a given total mass, the lowest value of  $f_0$  is obtained with the two masses equal, i.e.,  $m_1 = m_2$ . Increasing the total mass of the structure results in higher transmission loss values over the entire frequency range, as illustrated in Figure 6(a). Representing the changed values of  $f_0$  and  $f_b$  by  $f'_0$  and  $f'_b$ , the increase in transmission loss in the three frequency regions resulting from the total mass being increased from  $M$  to  $M'$  is as follows:

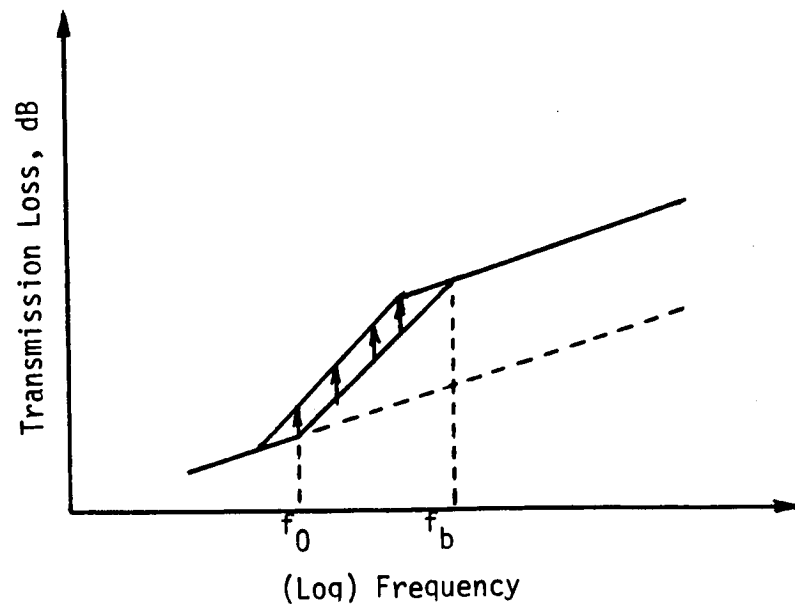
- o  $20 \log (M'/M)$  for  $f < f'_0$
- o  $40 \log (M'/M)$  for  $f_0 < f < f'_b$
- o  $20 \log (M'/M)$  for  $f > f_b$

The effect on the overall transmission loss of increasing mass for a panel exposed to the spectrum in Figure 1 is shown in Figure 7. The ordinate in this figure is the exterior A-weighted incident sound level, minus the transmission loss of the panel. It therefore represents the interior sound level without corrections for panel size and interior absorption. Note how influential the value of  $\Delta TL_M$  is on the interior A-weighted sound level. The lower line in the figure represents the theoretical minimum interior (uncorrected) levels that can be achieved from a simple double panel without bridges.

The effect of changing the panel separation  $d$  is less dramatic as illustrated in Figure 6(b). In this case, there is an increase in transmission loss of  $20 \log (d'/d)$  only in the frequency region between  $f_0$  and  $f_b$ , and this increase can never be greater than  $\Delta TL_M$ . Therefore, contrary to common belief, increasing the panel separation is only a useful method of increasing the overall transmission loss of a double panel when  $\Delta TL_M$  is high, and when the frequency range of interest lies between  $f_0$  and  $f_b$ .



(a) Effect of Increasing Panel Mass



(b) Effect of Increasing Panel Spacing

Figure 6. The Effects of Panel Mass and Separation on the Transmission Loss of a Double Panel

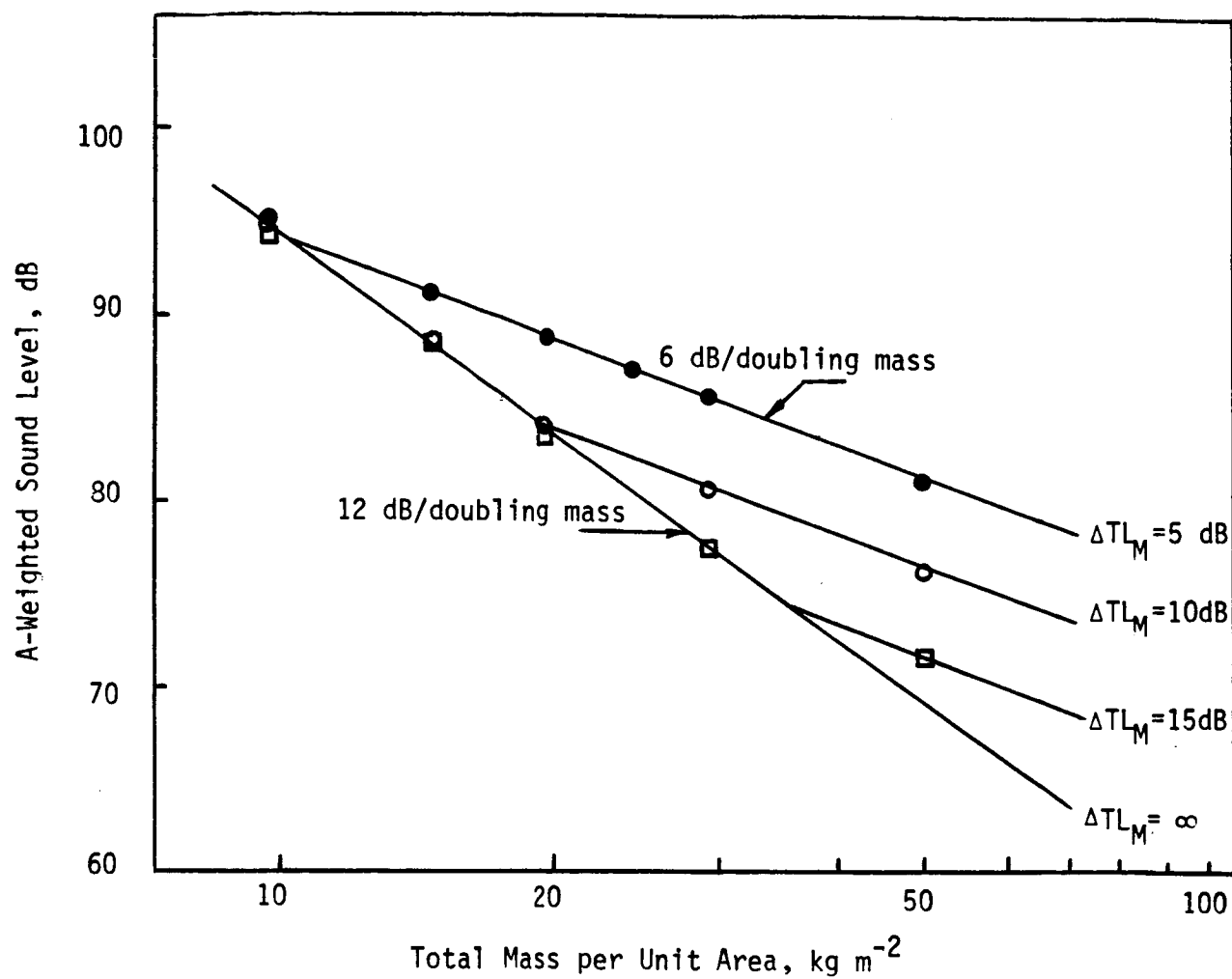


Figure 7. Interior Sound Level for Double Panel Sidewall



The final method of minimizing  $f_0$  is to reduce the values of density and the speed of sound in the cavity medium separating the panels. The actual increase in transmission loss  $\Delta TL$  only occurs at frequencies between  $f_0$  and  $f_b$ , and is equal to:

$$\Delta TL = 20 \log (\rho_2/\rho_2') + 40 \log (c_2/c_2') \quad (24)$$

Thus, changing the cavity pressurization from the equivalent of 1,500 m (5,000 ft) to 9,100 m (30,000 feet) would increase the transmission loss by 9 dB in this frequency region. For this increase to be achieved,  $\Delta TL_M$  must be at least 9 dB.

Methods of achieving the second goal, namely maximizing  $\Delta TL_M$ , can be identified by inspection of Equation (23).  $\Delta TL_M$  can be increased by 3 dB by doubling either the frame separation  $b$  or the panel critical frequency  $f_c$  (halving the panel material stiffness). Considerations of fuselage rigidity will normally eliminate the first option from contention, and the second as well if the critical frequency of both panels must be increased. However, similar effects can be obtained if the critical frequency of just one of the panels (the interior) is reduced. Moreover, it has been shown by Sharp<sup>4</sup> that higher values of  $\Delta TL_M$  can be obtained by connecting the two panels together by means of points applied to the framework supporting one of the panels. If the two panels are of equal mass, and the points are regularly spaced on a square lattice of size  $e$  meters, then  $\Delta TL_M$  can be obtained from the following expression:

$$\Delta TL_M = 20 \log (e f_c) - 51, \text{ dB} \quad (25)$$

Note that the factor "20" in the first term of this equation effectively doubles the increase in  $\Delta TL_M$  that can be achieved by increasing  $f_c$  compared to the line frame connections assumed in Equation (23). A comparison of the values of  $\Delta TL_M$  for line and point bridges between panels is given in Figure 8.

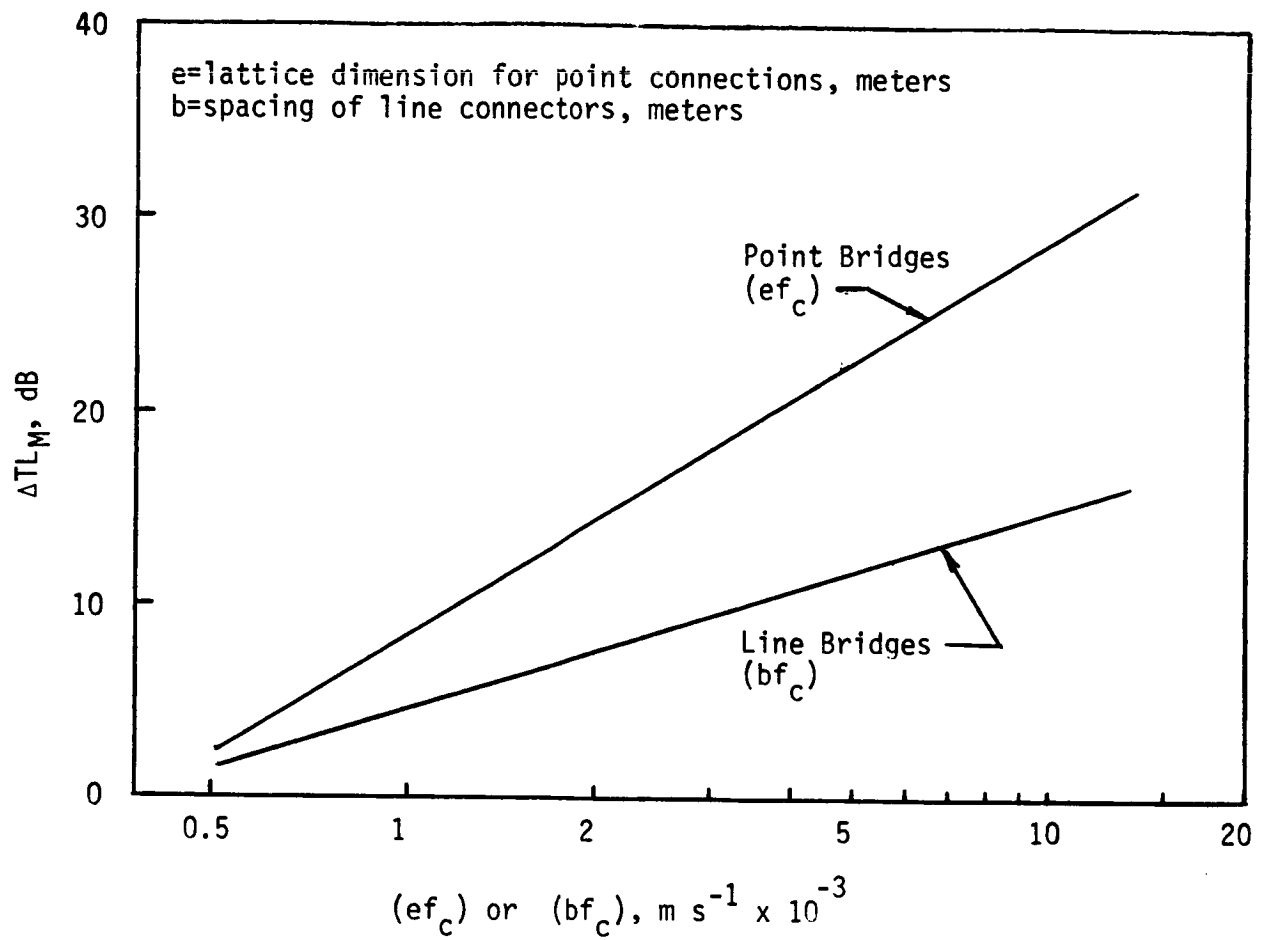


Figure 8. Values of  $\Delta TL_m$  as a Function of  $ef_c$  and  $bf_c$

### 2.3.3 Cavity Absorption

At the start of this section on double panels, it was stated that absorption was required in the cavity to obtain the full benefits of the construction. It has been shown by Sharp<sup>4</sup> that without cavity absorption, the transmission loss of a double panel is often no better than that which would be obtained from a single panel of the same total mass. In other words the increase in transmission loss at frequencies greater than  $f_0$  is not achieved. The reason for this is the formation of lateral acoustic modes across the width of the panel, i.e., between the frames, so that the entrapped air no longer behaves as a stiffness element - a requirement upon which the double-panel isolation effect depends. Thus the theoretical values of transmission loss at frequencies greater than  $f_0$ , as shown in Figure 5, are obtained only up to the frequency of the first lateral cavity mode.

This result provides an interesting method by which the transmission loss of double panels can be maintained without the use of absorptive material. If the cavity is divided into a large number of small cavities by means of a lattice network, the entrapped air will behave as a stiffness element up to high frequencies, i.e., up to the lateral modal frequencies of the individual elements in the lattice. This is demonstrated in the measured results of Figure 9, where the lattice dimension is 0.61 m (2 ft) square. At low frequencies, the measured results follow the predicted curve closely. The strong coupling effect of the first and second lateral modes of the lattice (in the 315 Hz and 630 Hz one-third octave bands) is evident. The lattice has very little effect at high frequencies. If the lattice dimensions were 0.15 m (6 in) rather than 0.61 m, it is anticipated that the predicted results would be approached at all frequencies up to 1000 Hz without the use of any absorption material.

### 2.3.4 Triple Panels

The possibility of obtaining transmission loss values in excess of the calculated mass law has been demonstrated in the discussion on double-panel constructions. In an attempt to obtain even greater values of transmission loss from a construction, it is a natural extension to study the acoustical

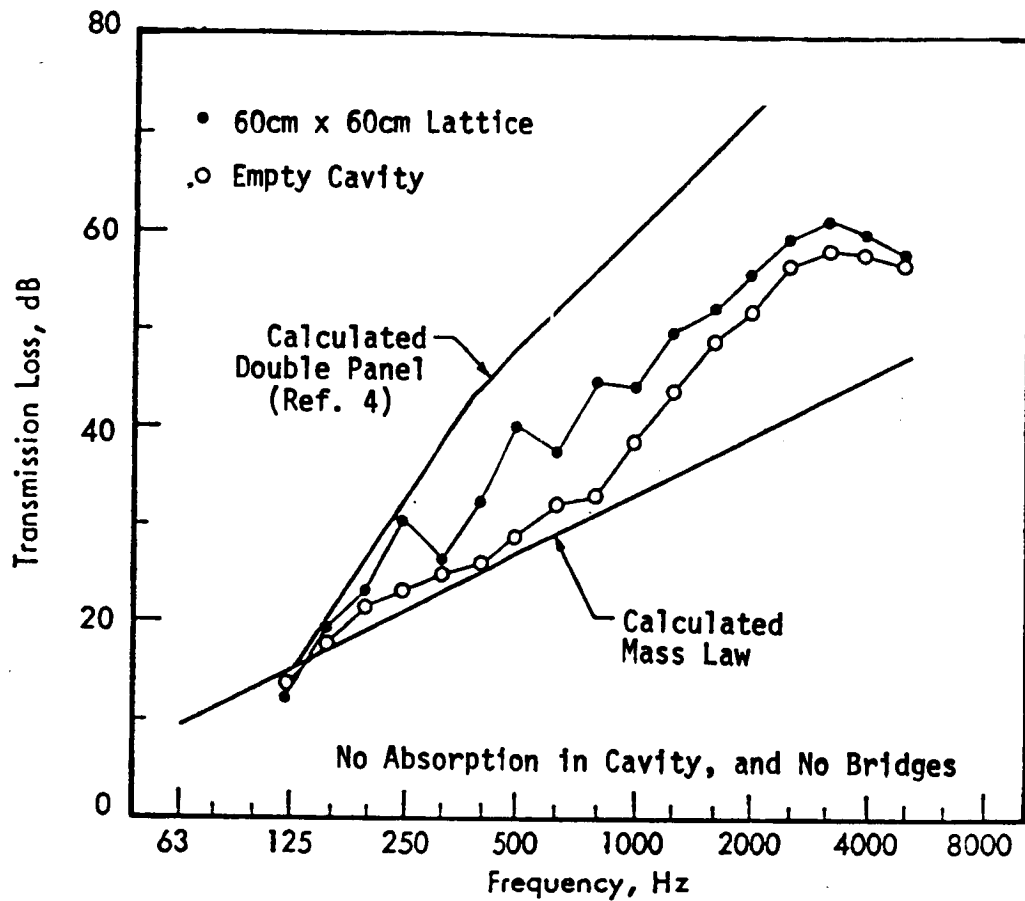


Figure 9. Measured Values of Transmission Loss of an Isolated Double Panel Construction with a 60 cm by 60 cm Lattice in the Cavity. The construction consists of 6.4 mm and 3.2 mm hardboard with a spacing of 16 cm.

characteristics of triple panels. The general principles are just the same as those described in the previous section and, not surprisingly, the results prove to be similar. In this case, however, there are two resonant frequencies  $f_+$  and  $f_-$  in place of the single resonance at  $f_0$  for the double panel. At frequencies less than  $f_0'$ , where

$$f_0' = \sqrt{f_+ f_-}$$

the transmission loss is given by the mass law as stated in Equation (5) where  $m = M$ , the total mass of the structure. Above  $f_0'$ , there is a dip or a flattening of the curve at  $f_+$ , but provided that there is absorption in the cavity, the transmission loss increases at a rate of 30 dB per octave up to the frequency  $f_b$  where transmission through the panel connections becomes the major path through the structure.

To obtain maximum benefits from a triple panel, it is necessary to select the parameters such that the higher of the two resonant frequencies,  $f_+$ , is below the frequency range of interest. If the media in the two cavities are identical, then the lowest value of  $f_+$  for a triple panel of given overall mass and thickness occurs with the three panels equally spaced and with a mass distribution in the ratio 1:2:1, i.e., the center panel being twice the mass of each of the two outside panels. With these optimum parameters, the relationship between the transmission loss of double and triple panels, of the same overall mass and thickness, is shown in Figure 10. The value of the frequency  $f_0'$  for the triple panel is equal to  $\sqrt{3} f_0$ , and the two curves cross at a frequency equal to  $\sqrt{8} f_0$ , where the transmission loss is 18 dB greater than the value given by the simple mass-law. Thus the triple panel is an improvement over the double panel only at frequencies greater than  $\sqrt{8} f_0$ , and then only if the value of  $\Delta TL_M$  is greater than 18 dB.

The performance of a triple panel can be improved by changing the characteristics of the media in one or both cavities. In an aircraft structure, this could be achieved conceptually by venting the outermost cavity to the exterior atmosphere, assumed to be 9,100 m for cruise altitude. Figure 11 shows the effect of such a change in medium on the quantity

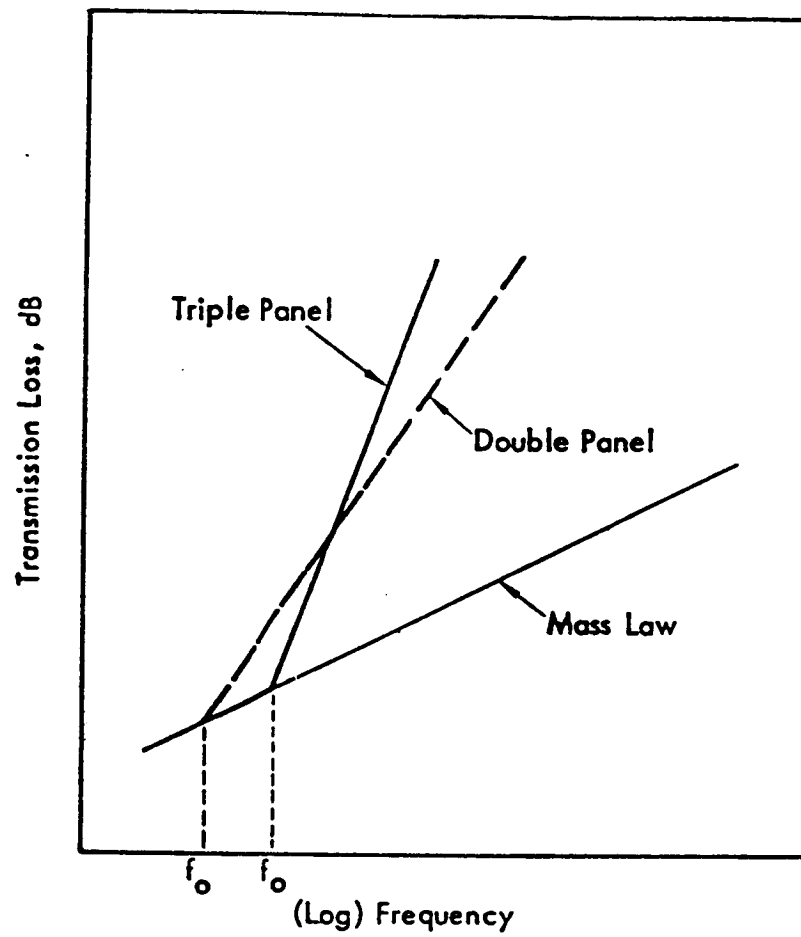


Figure 10. A Comparison of the Transmission Loss Provided by Double and Triple Panel Constructions of Equal Total Mass and Overall Thickness with no Sound Bridges.

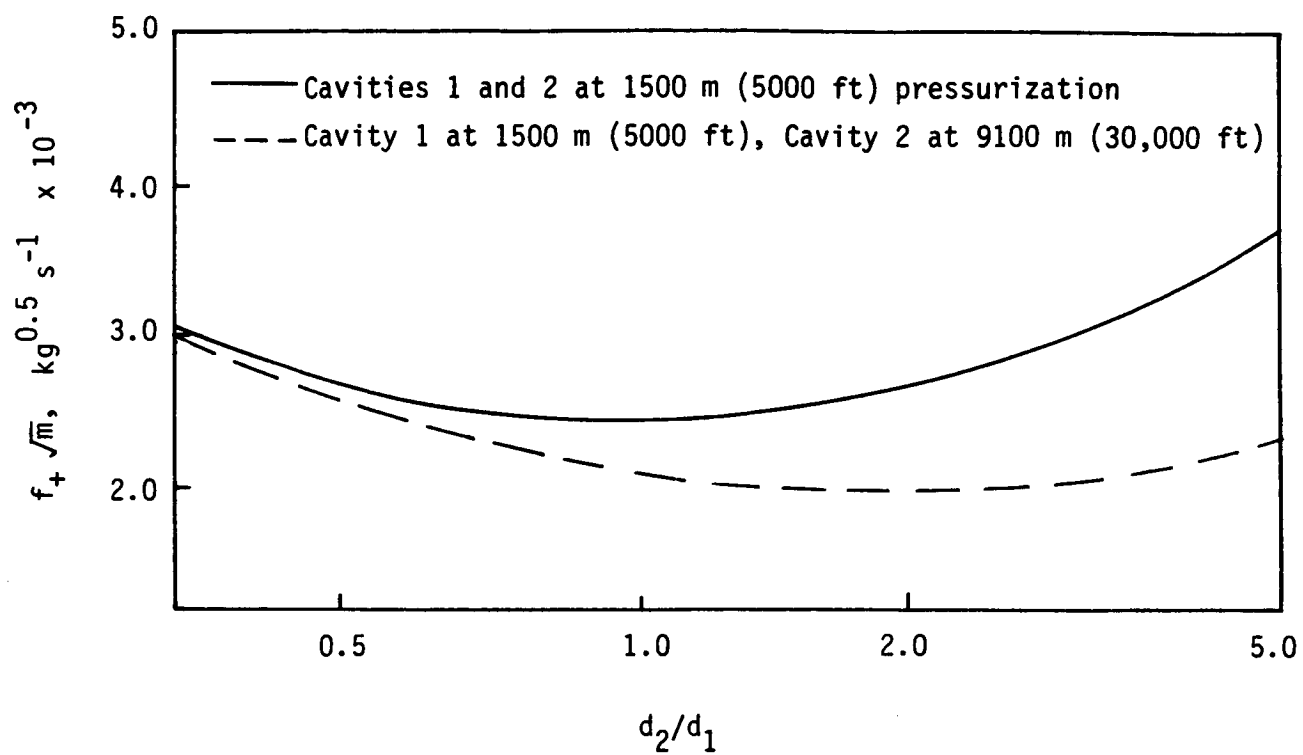


Figure 11. Effect of Cavity Medium Density on Triple Panel Resonance.

$f_+ \sqrt{m}$ , where  $m$  is the mass of the outer leaves of the triple panel (assumed to be of 1:2:1 configuration), as a function of the ratio of cavity dimensions. Note that the optimum condition for lowest  $f_+$  does not occur with equal cavity dimensions (i.e.,  $d_2/d_1 = 1$ ) when the cavity media are different. The minimum value of the quantity  $f_+ \sqrt{m}$  is reduced by a factor of 1.22, and the value of  $f'_0$  is reduced by a factor of 1.3. The result is, of course, an improvement in transmission loss in the frequency region near  $f'_0$ . However, assuming realistic values of  $\Delta TL_M$  in the range 10 to 15 dB, the triple panel with one cavity at low pressure is only marginally better than an equivalent double panel with a cavity pressure equal to the cabin pressurization. Accordingly, it is concluded that triple panels do not offer any advantages for this program.

## 2.4 Panel Damping

A standard approach to reducing panel vibration levels, and hence increasing transmission loss, is by the application of damping material. In non-aircraft applications, damping materials are often combined with added mass. Weight considerations require that aircraft applications of damping material be accomplished in an efficient manner, so that maximum benefit is achieved from a given mass of damping material. An innovative approach to the application of damping is the method of intrinsic tuning and damping<sup>6</sup> which has been developed for intermediate- to low-frequency vibration control of periodically stiffened structures. It is a design method aimed at spoiling dominant modes, and providing a route to drain vibratory energy from the structure. The concept is illustrated in Figure 12.

Consider a panel periodically supported on rigid stiffeners. A principal mode in response to a normal plane wave is sketched in Figure 12(a). Each skin bay responds at a fundamental frequency  $f_{\text{panel}}$ , determined by the rigid boundary condition at each stiffener. The stiffeners have resonant frequency  $f_{\text{stiffener}}$  much higher than  $f_{\text{panel}}$ . If the stiffeners and/or skin are modified such that  $f_{\text{stiffener}} = f_{\text{panel}}$ , the mode shown in Figure 12(a) does not exist; the stiffeners no longer provide the dynamic stiffness boundary conditions necessary to support the mode. Two other principal modes now exist: an in-phase mode at a frequency below  $f_{\text{panel}}$  (Figure 12(b)) and an



out-of-phase mode at a higher frequency (Figure 12(c)). In the absence of damping, it might be supposed that no net benefit would be achieved: mode (a) is spoiled, but is replaced by (b) and (c). However, in the modes shown in Figures 12(b) and 12(c), the stiffeners are moving. The strains at the edges of the stiffeners are considerably larger than on the skin, due to the frame height. A constrained damping layer properly placed on the stiffeners might therefore be highly effective in controlling the motion.

Experiments described in Reference 6 showed that noise and vibration levels were reduced by up to 6 dB by tuning. Benefits were seen at the tuned frequency and at higher frequencies: because the technique couples the structure to provide a path to effectively placed damping, the benefit is not just at a single frequency as with simple mechanical resonators. This approach is based on modification of the existing structure, rather than by adding mass, and therefore is attractive for aircraft. Substantial modification of the structure is required, however, to match the panel and stiffener frequencies. Tuned panels in Reference 6 had stiffener spacings half as great as in typical passenger aircraft. An intrinsically tuned panel is therefore likely to be structurally less efficient than one designed without tuning.

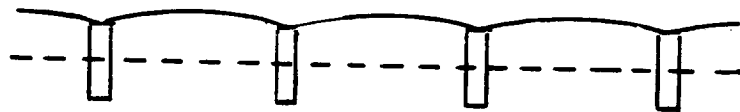
In the present study, intrinsic tuning was evaluated by using a test panel with stiffener spacing reduced so as to match skin bay and stiffener fundamentals. Skin and stringer properties are otherwise the same as for a "normal" aircraft fuselage. If these results compare favorably with other techniques, a future program should investigate a more efficiently designed, intrinsically tuned structure.

## 2.5 Resonator Panels

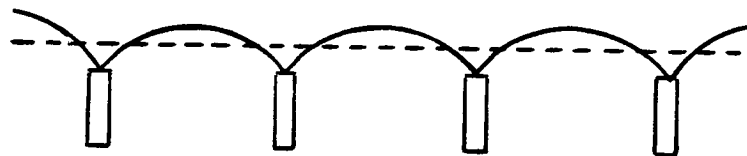
The spectrum of the sound incident on the exterior fuselage, as shown in Figure 1, consists of a series of discrete amplitude spikes at harmonically related frequencies, the fundamental being 160 Hz. Methods of achieving high transmission loss that have been reviewed in this section have relied upon physical parameters such as mass, thickness, and stiffness, which provide a fairly uniform relationship with frequency. An alternative approach is to



(a) Fundamental Mode of Skin on Rigid Stiffeners



(b) In-Phase Mode, Tuned Structure



(c) Out-of-Phase Mode, Tuned Structure

Figure 12. Principles of Intrinsic Tuning.

provide high values of transmission loss only at the frequencies where the exterior level is greatest. This can be achieved by the use of resonant systems built-in to otherwise conventional structures. A promising technique is to use an acoustic resonator, with the resonator formed to a large extent by existing volume elements in the structure.

The acoustical resonator consists of a small mass of air acting on the stiffness of an enclosed volume of air to provide a resonance. The most often cited example of an acoustic resonator consists of a bottle with a thin neck - the air in the neck representing the mass, and the air in the main body of the bottle the stiffness. If the absorption in the system is low, then sound will be radiated by the opening in the neck at the resonant frequency. If the absorption is high, then sound will be absorbed. Westphal<sup>7</sup> has shown that the transmission loss of a honeycomb panel can be increased by up to 10 dB by drilling holes in one of the face plates - one hole per honeycomb cell. Postlethwaite<sup>8</sup> has also shown similar results over narrow frequency ranges.

The principle of operation can best be understood by reference to the equivalent electrical circuit for a structure consisting of a number of resonators attached to a single panel - see Figure 13. Resonance occurs at a frequency given by:

$$f_{\text{res}} = \frac{1}{2\pi} \sqrt{\frac{k}{m'}}$$

where  $k$  is the stiffness of the air in the volume of the resonator, and  $m'$  is the mass of air in the "neck" of the resonator, corrected for end effects.<sup>9</sup> At this frequency, sound radiated by the openings of the resonators, represented by the resistor " $r$ ", is out of phase with the sound radiated by the main panel into the characteristic impedance  $\rho c$ , thus leading to a net decrease in sound radiated, and hence an increase in transmission loss. Using the analogous electrical circuit shown in Figure 13, the calculated effect of the resonators on the transmission loss of the base panel is given in Figure 14, showing the considerable increase at 400 Hz, the resonant frequency chosen for this example. The improvement in transmission loss and the bandwidth over which it occurs are dependent on the damping in the resonator.

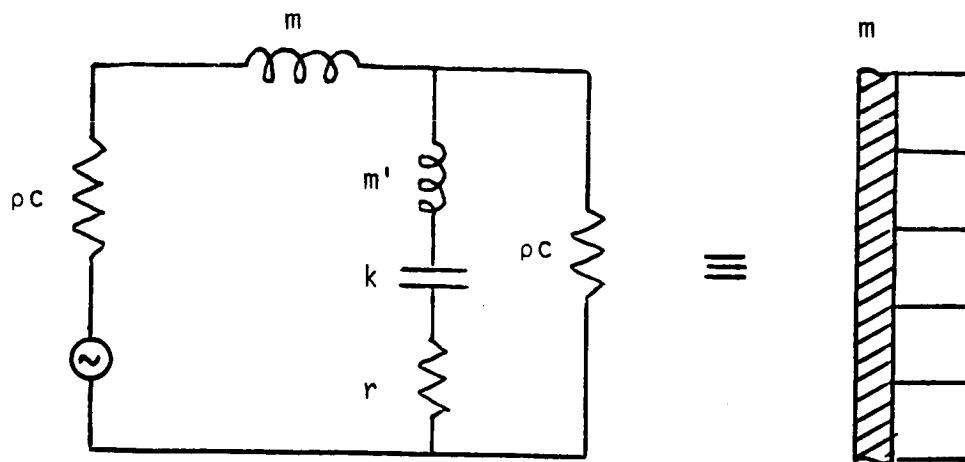


Figure 13. Equivalent Electrical Circuit Representation of a Resonator Panel

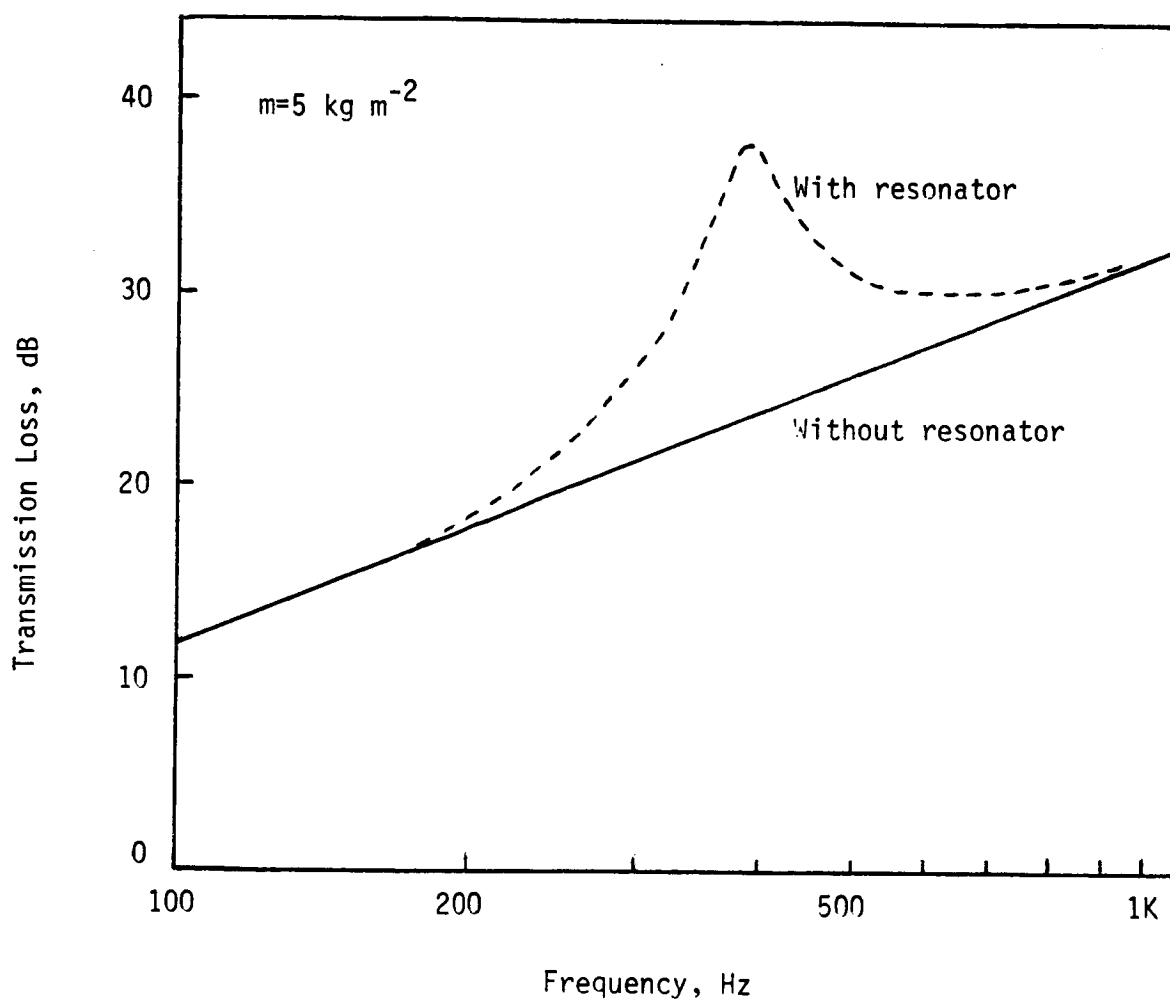


Figure 14. Transmission Loss of a Single Panel with and without a Resonator Tuned to 400 Hz.

## 2.6 Summary

A review of the results obtained from the simple analyses described earlier in Section 2 provides the following conclusions:

### Single Panels

- o For typical aircraft exterior skins, the mass-law is valid for most of the frequency range of interest in this study.
- o The transmission loss of single panels increases at frequencies less than  $f_{11}$ . Therefore, designing for a high value of  $f_{11}$  may prove to be a useful method of improving performance.
- o Increasing the transmission loss over the entire frequency range by designing for a very low critical frequency does not appear possible with existing core materials.
- o Sandwich panels can be designed to exhibit high static stiffness and high critical frequency by careful design of the core. Such panels are useful when the panel itself must provide the necessary static rigidity. In aircraft structures, the frames and longerons provide the stiffness. Thus the use of such panels is unnecessary.

### Multiple Panels

- o Double panels can have a higher transmission loss than single panels at frequencies greater than the fundamental resonance at  $f_0$ . At higher frequencies, the transmission loss is determined largely by the quantity  $\Delta TL_M$  which is a function of the number and type of panel connections and the flexibility of the panels. Overall increases in transmission loss over a wide frequency range require a reduction in  $f_0$  and an increase in  $\Delta TL_M$ .

- o The frequency  $f_0$  at which the fundamental resonance occurs can be reduced without changing the overall mass or dimensions by venting the cavity to reduce the cavity stiffness, or by reducing the cavity pressure.
- o The value of  $\Delta TL_M$  can be increased by using point spaces for connection of the inner trim panel to the main frame. Further increases are possible by incorporating resilient material for these point spacers.
- o Absorption in the cavity necessary to achieve the full double-wall effect can be achieved without the use of excessive amounts of acoustic material by incorporating a latticework to break up lateral acoustic modes.
- o For the same overall mass and thickness, a double-panel is generally superior to a triple-panel structure. The added complications of designing and constructing a triple-panel structure are not worth the few benefits that may be obtained.

### Resonant Structures

- o Acoustical resonators incorporated into a single- or multiple-panel designs appear to have the capability of increasing the transmission loss over a narrow frequency range.
- o The application of tuned damping, where the resonant frequencies of the panels and frames are designed to be similar, may have application to turboprop fuselage design.

## 3.0 TEST PANEL DESIGNS

The objective of this study was to develop new designs for turboprop aircraft sidewalls providing higher values of noise reduction at frequencies specific to the turboprop sound signature. As a result of the analyses in the previous section, individual techniques and combinations of several techniques

were incorporated into the design of seven test panels suitable for laboratory testing. The purpose of these designs is to demonstrate principles that, if successful, can be translated into structures suitable for aircraft application. Accordingly, for cost purposes the test panels were flat rather than curved and were not necessarily constructed of aircraft-suitable materials, nor were they specifically intended to provide an interior sound level of 80 dBA for the exterior sound spectrum shown in Figure 1. The test panel designs are detailed in Appendix A. The panel types are denoted A through K; the several configurations of each panel are designated A.1, A.2, B.1, B.2, B.3 and so on.

### 3.1 Panel A - Double Wall With Vented Cavity

Panel A was designed to test the hypothesis that reducing the stiffness of the air in the cavity of a double panel by providing vents to the outside (i.e., to the underfloor or ceiling regions of an aircraft) will reduce the value of the fundamental resonant frequency  $f_0$ . According to Figure 6(b), the effect of reducing  $f_0$  at constant overall mass, provides an increase in transmission loss up to the frequency  $f_b$ . In Panel A,  $\Delta TL_M$ , and hence  $f_b$ , are kept high by minimizing the number of connections between two panels of hardboard - see Figure A-1 - thus separating out the effects of bridging and coincidence and allowing a review only of the effect of venting on the value of  $f_0$ . Panel configurations with and without absorption in the cavity were specified. For comparison, a non-vented version (Panel B.1) was specified to have a sealing tape at the edges of the cavity, which was not expected to significantly increase the contacts between the two panels. To prevent flanking transmission around the open edges of the vented panel from contributing to the received sound signal, the sheet on the source side was made sufficiently large for transmission via this path to lie outside the time window employed in the time delay spectrometry test method described in Section 4.0.

### 3.2 Panel B - Double Wall With Cavity Lattice

Panel B was designed to test the hypothesis that the effects of acoustic absorption in the cavity can be obtained without acoustic absorption material

by breaking up the lateral cavity modes using a square latticework - see Figures A-2 and A-3. In this design, the spacing of the lattice is 15.2 cm (6.0 in), so that the transmission loss of the construction should follow the theoretical double-panel curve up to about 1000 Hz.

### 3.3 Panel C - Wall Resonator

Panel C was designed to test the performance of a single panel with attached resonators with the objective of achieving increased transmission loss over a narrow band of frequencies. The base panel was constructed of 1.27 cm (0.5 in) plywood, to which were attached 2.5 cm (1 in) long sections of 7.6 cm (3 in) diameter cardboard tubes - see Figures A-4, A-5 and A-6. Covering the other surface of the tubes is a 6.4 cm (0.25 in) sheet of plexiglass. The resonators were formed by drilling holes in the plexiglass at the center of each cardboard tube. With this arrangement, resonant frequencies were predicted to occur as follows:

<u>Hole Diameter</u>	<u>Resonant Frequency</u>
3.2 mm (0.125 in)	155 Hz
6.4 mm (0.25 in)	275 Hz
12.7 mm (0.5 in)	460 Hz

In addition, 1.9 cm (0.75 in) long tubes were inserted into the 6.4 mm diameter holes to reduce the resonant frequency from 275 Hz to 185 Hz.

### 3.4 Panel D - Double Wall with Reduced Cavity Pressure

The ultimate application of reducing sidewall cavity pressure might involve venting the cavity to ambient, resulting in an aircraft with a pressure wall located at the current trim panel and in an unpressurized cavity. Fully testing such a structure in a laboratory would require partially evacuating the source room and test panel cavity. However an indication of the acoustical performance of such a sidewall can be gained by depressurizing only the cavity in order to determine whether the measured noise reduction through such a sidewall conforms to expectations from theory.



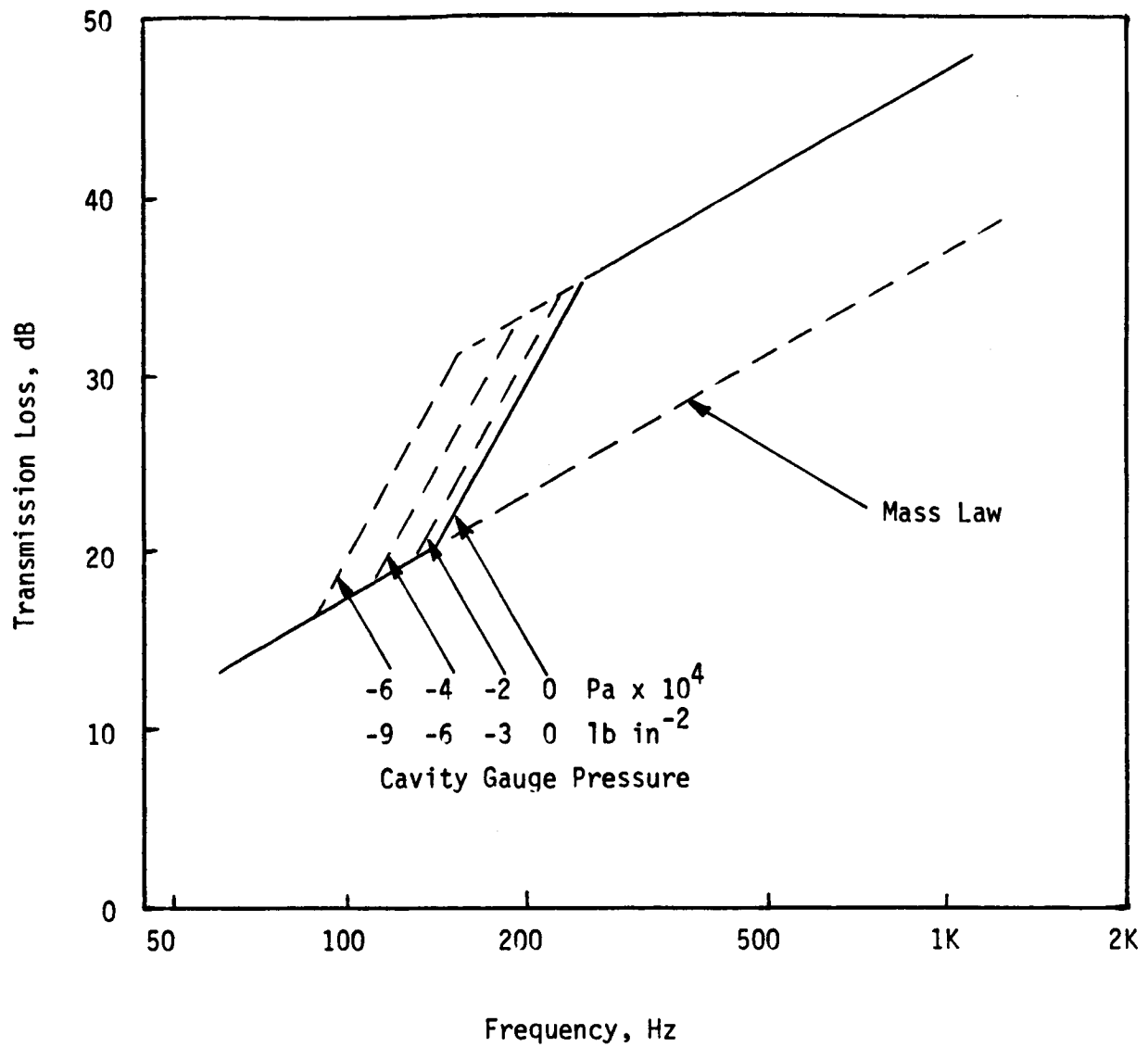


Figure 15. Predicted Transmission Loss for Panel D for Various Pressure Reductions of the Cavity (Referenced to the Pressure Outside the Cavity during Testing)

Accordingly Panel D was derived from a typical aircraft fuselage sidewall (primary structure only) consisting of a 1.6 mm (0.063 in) aluminum skin on a standard framework - see Figures A-7 and A-8. A second skin of 1.6 mm aluminum was attached to the frames and separated from the first skin by means of point connectors or spacers. Tests were specified with the cavity between the two panels evacuated to various gauge pressures up to as high as  $6.2 \times 10^4$  Pa (9.0 lb/in<sup>2</sup>). The values of the fundamental resonance,  $f_0$  equal to 142 Hz (for normal incidence) at sea level, were predicted to be reduced such that, with point connectors included to increase the value of  $TL_M$  to on the order of 12 dB, the predicted values of transmission loss were as shown in Figure 15.

### 3.5 Panel E - Skin-Longeron Tuning

Panel E (see Figures A-9, A-10 and A-11) was included in the series of test panels to study the effectiveness of intrinsic or skin-longeron tuning and damping. For cost purposes, the panel was derived from conventional aircraft sidewall primary structure modified for the test by adjusting longeron pitch and mass, as described below, such that the longerons and the panel bays between them were either tuned, i.e., had a similar resonant frequency, or were untuned. (A production sidewall built to this principle would be tuned in design rather than by the process described here.) The effect of damping material was then to be assessed for both tuned and untuned conditions, on the basis that it would be more effective in the tuned condition by dissipating vibrational energy transferred from the panel bays to the longerons.

Damping applied to this panel was in the form of a high temperature fused PVC alloy sheet (EAR Isodamp C-2003-050) applied to the longeron flange using an adhesive and constrained by a strip of aluminum. The thickness including the constraining layer was 2.87 mm (0.113 in). In order to prevent the addition of damping material to the longeron flanges from adding mass that could affect the tuning, lead of identical mass to the damping material [0.06 kg/m (0.04 lb/ft)], was applied to the longeron flanges during panel tuning and was retained there when testing the panels in their "undamped" configurations.

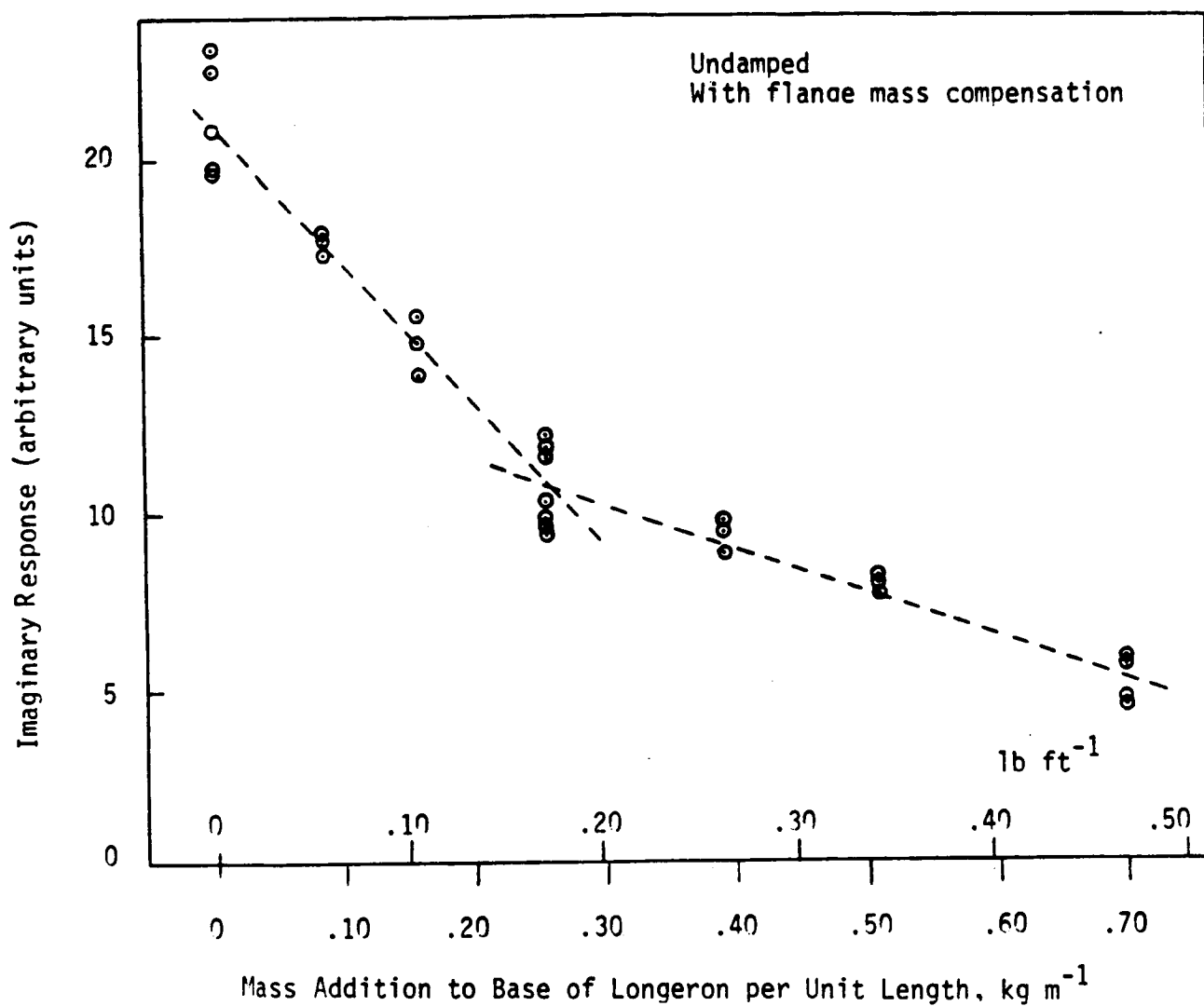


Figure 16. Determination of Added Mass Necessary to Tune Panel E

The aluminum panel had twice as many longerons as the aircraft panel from which it was derived, in order to bring the resonant frequencies of the longerons and skin bays into the same general range. Actual tuning of the panel was then achieved by adding lead uniformly along the base of the longerons surrounding several skin bays in the center of the panel and observing the impact response of the center skin bay as a function of added mass. The response of the panel can be expected to be a superposition of two influences: a steadily decreasing response due to the inertia of the added mass; and an abrupt change in response for the mass at the tuned condition. The resultant curve should then evidence the tuned condition by a discontinuity, at the very least, in the decreasing response due to addition of mass.

The test method is illustrated in Figure A-10 and the results are shown in Figure 16, in which each response is the average of three impacts and was repeated at least three times. The resonant frequency was determined in this process to lie in the range 369 to 378 Hz (depending on longeron mass), which agreed well with calculations. The longeron mass required to achieve tuning also agreed well with predicted values. The added mass value was 0.26 kg/m (0.17 lb/ft).

The following test configurations for Panel E were then defined:

E.1: the bare panel - no added mass, no damping\*

Untuned Condition A

E.2: less mass than required for tuning, not damped

E.3: less mass than required for tuning, damped

} Same total  
mass

Tuned Condition

E.4: mass as required for tuning, not damped

E.5: mass as required for tuning, damped

} Same total  
mass

Untuned Condition B

E.6: more mass than required for tuning, not damped

E.7: more mass than required for tuning, damped

} Same total  
mass

\*This is the same panel as K.1

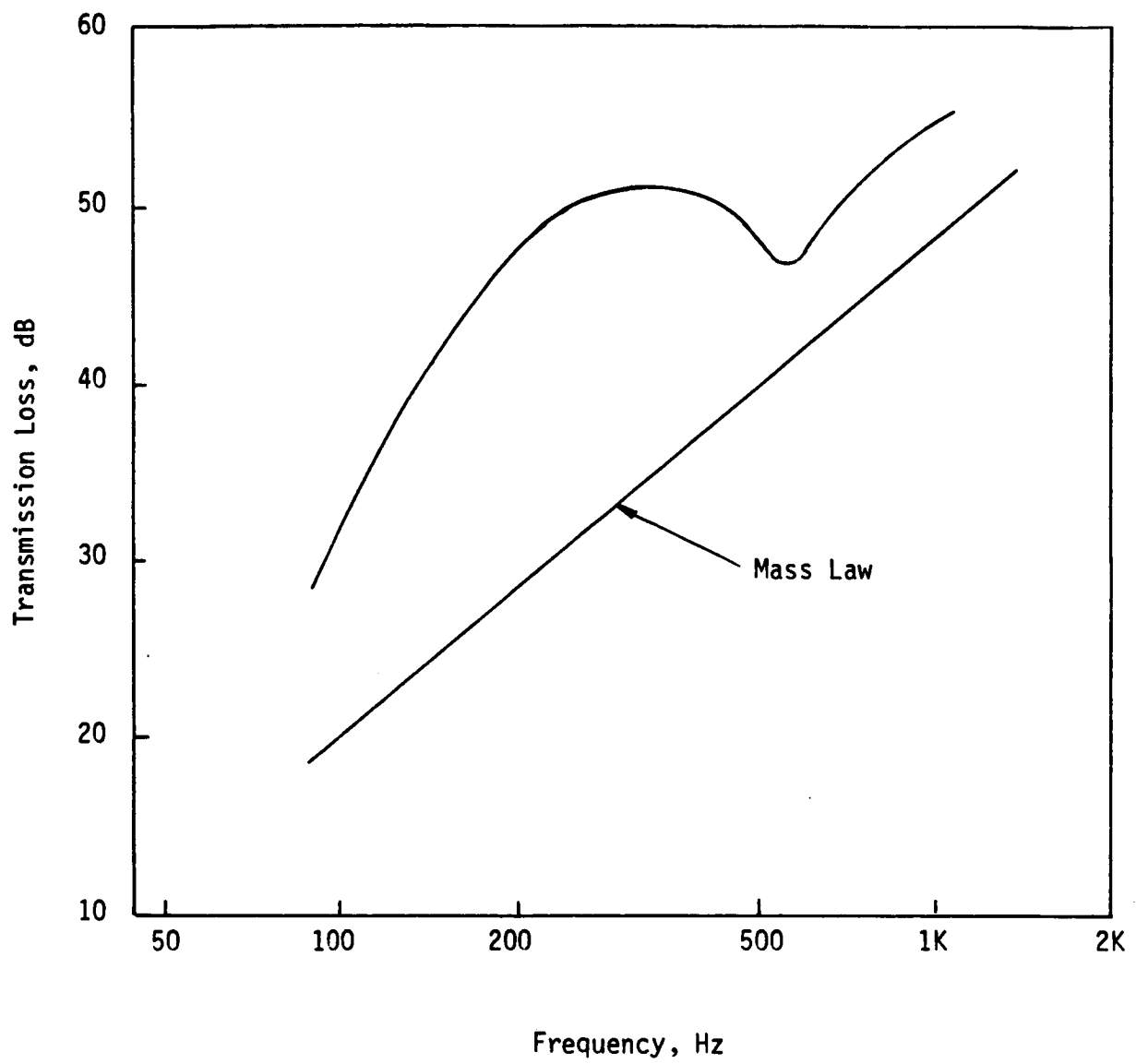


Figure 17. Predicted Transmission Loss for Panel F.

### 3.6 Panel F - Double Wall With Stiff Panel

Panel F was the combination of a stiff panel and a limp panel in a double wall design with point connectors separating the limp panel from the rest of the structure. The exterior skin was a honeycomb panel, which when firmly attached to the frames, has a fundamental resonant frequency  $f_{11}$  of about 600 Hz, thus providing high transmission loss between the fundamental double-panel frequency  $f_0$  (of about 130 Hz) and 600 Hz due to the panel stiffness - see Figures A-12 and A-13. The predicted value of transmission loss is shown in Figure 17.

### 3.7 Panel G - Double Wall With Resonator

Panel G was the combination of a panel with resonators in a double wall design - see Figures A-14 and A-15. The resonators were tuned to a frequency predicted to be about 190 Hz, so as to provide increased transmission loss in the frequency region immediately above the fundamental double-panel frequency  $f_0$  of 130 Hz. To examine the effectiveness of this type of structure without the added complications of sound bridges, the second panel in this design was a  $4.6 \text{ kg/m}^2$  ( $0.93 \text{ lb/ft}^2$ ) flexible "loaded" vinyl sheet.

### 3.8 Panel H - Conventional Sidewall 1

This panel (see Figures A-16 and A-17) is representative of conventional, turbofan aircraft sidewall primary structure, and constitutes a contractor-supplied baseline or reference panel for comparison with the other "research" panels. The two configurations test the effect of reversing the panel in the test window, since some of the other panels were scheduled for test "reversed" in order to facilitate configuration changes.

### 3.9 Panel I - Conventional Sidewall 2

Figure A-18 presents details of another contractor-supplied baseline or reference panel identical to Panel H except for an increased skin thickness which is representative of the skin thickness to resist acoustic fatigue in the study aircraft environment.

### 3.10 Panel J - Skin Only

This panel (see Figure A-19) consists of a single, unstiffened skin of thickness identical to Panel I.

### 3.11 Panel K - Conventional Sidewall - Close Longeron Pitch

In this panel, longeron spacing is half that of Panel H to change the natural frequency and to permit comparison with Panel E. See Figure A-20.

#### 4.0 ACOUSTICAL TESTING

To minimize cost, and in view of the exploratory nature of the program which called for comparison of the acoustical performance of many panel configurations, a simple, laboratory testing technique was selected - as well as the use of the flat, comparatively small test panels described earlier. Although the modal response of the fuselage ultimately needs consideration in sidewall design against turboprop noise, considerable information on treatment effectiveness can be derived from a simple noise reduction test on 1.2 m by 1.8 m (4 ft by 6 ft) panels which are of the same dimensional order as floor-to-bag rack portions of sidewall extending for two to three frame bays.

The simple test method employed was a time delay spectrometry (Ref. 10) (TDS) technique in which a short duration signal is used as the source rather than a traditional, continuous noise signal. In principle, by employing suitable source-panel-microphone geometries and an appropriate time window for analyzing the transmitted sound signal, the TDS method can be used to exclude flanking transmission and thus allow transmission loss testing of free, i.e., unmounted, panels, or of individual components of structures in their "real-life" or field installations. However, for practical-size panels and for testing down to 100 Hz, the time settings of the system cannot be adjusted to exclude flanking transmission without compromising frequency resolution. The panels were therefore mounted in a window between two acoustic rooms. Although this environment also lends itself to conventional transmission loss testing, the TDS technique was employed for its good frequency resolution, and its high signal to noise ratio and dynamic range.

Figure 18 shows a comparison of noise reduction as measured by the TDS technique employed here and by a continuous noise technique involving the same source-panel-microphone geometry in the same acoustic rooms. The TDS technique permits transmission studies over a continuous range of discrete frequencies, an advantage, while the broadband continuous noise test result also shown in Figure 18 used one-third octave analysis. Exact equivalence of results from the two methods is not, therefore, expected.



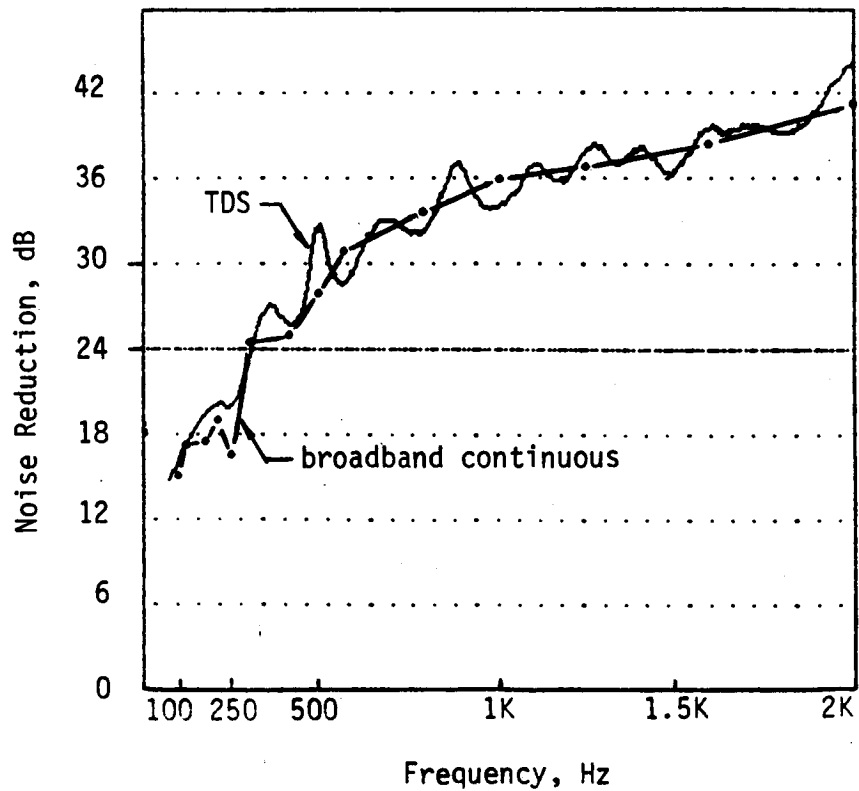
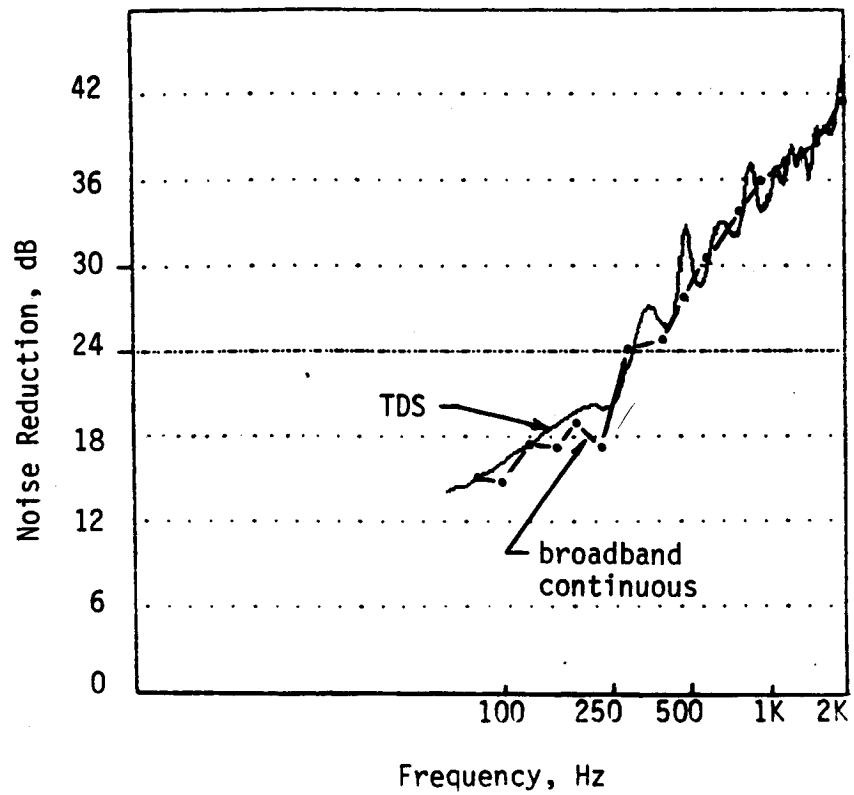


Figure 18. Noise Reduction vs. Frequency (Log and Linear Scales) for a Conventional Aircraft Sidewall without Trim (Panel J in Appendix A).

#### 4.1 Test Facility and Instrumentation

Figure 19 illustrates the noise reduction test facility and the locations of source and receiving instrumentation that were used for determining noise reduction. The test panels were clamped and sealed to the source side of the window. For panels having two walls, the wall on the source side was rigidly clamped to the window (except for Panel A where the wall on the receiver side was clamped).

Table 1 describes the facility dimensions and source-panel-receiver spacing.

TABLE 1.

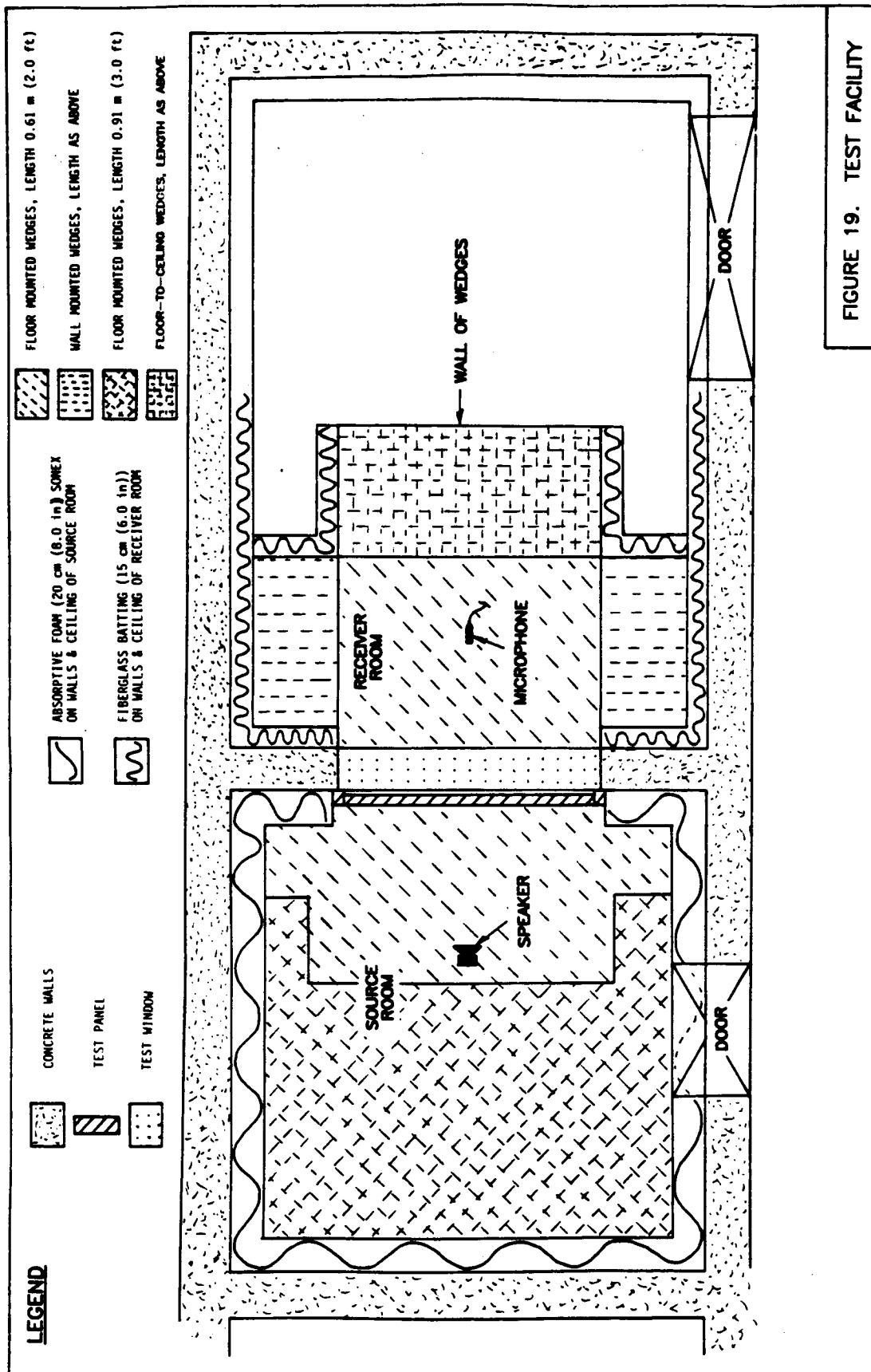
Dimensions of the Noise Reduction Test Facility. Wedge tip to wedge tip dimensions are indicated where asterisked.

Source Room*	2.9 m x 2.9 m x 1.4 m (9.5 ft x 9.5 ft x 4.5 ft) 11.5 m <sup>3</sup> (406 ft <sup>3</sup> )
Receiving Room*	1.2 m x 1.8 m x 1.5 m (4 ft x 6 ft x 5 ft)
Test Window	1.2 m x 1.8 m x 30 cm (4 ft x 6 ft x 12 in)
Source-Panel Distance	1.0 m (3.3 ft)
Panel-Microphone Distance	1.1 m (3.7 ft)

A block diagram of the instrumentation is shown in Figure 20.

#### 4.2 Measurement and Analysis Method

After calibration of the microphone and sound source system, measurements were taken without a test panel installed to obtain energy-time curves. This was achieved by generating a sinusoidal test signal swept through the frequency range and analyzed when received by the microphone via the tracking filter of the analyzer. Quick-look analysis of the energy time curve was made in order to determine the appropriate time delay offset for acquisition of frequency response data. The frequency response was then acquired over the frequency range of 4 Hz to 2000 Hz with a tracking filter bandwidth of 18 Hz. Similar measurements were then taken with each test panel installed in the test window. These data were stored on floppy disk for later reduction to



noise reduction plots by differencing the frequency response curves with and without a test panel installed.

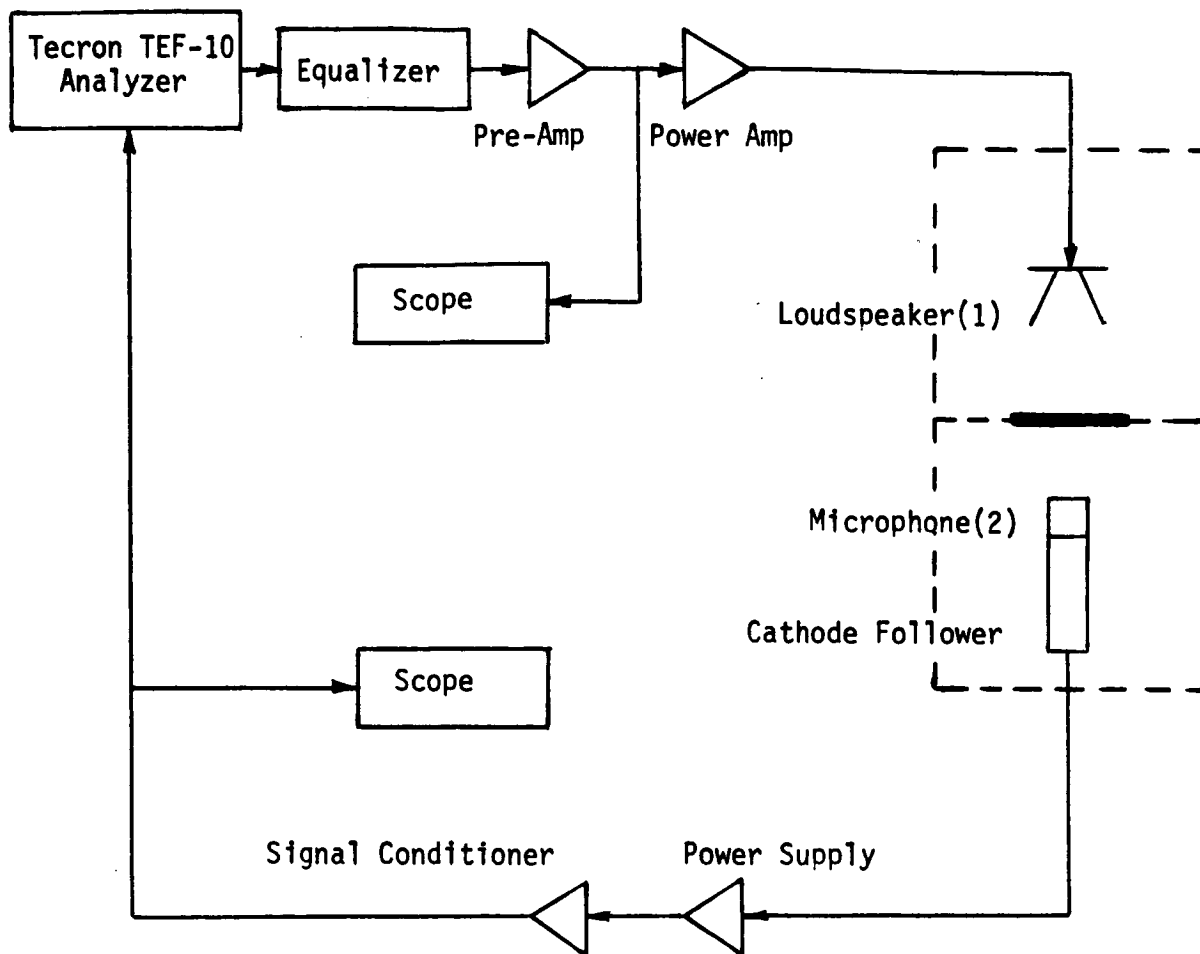
Throughout the test, quick-look repeat measurements were taken to ensure repeatability of the data, and re-tests were performed as required.

#### 4.3 Noise Reduction Data

In this program the acoustical performance of the panels is defined in terms of "noise reduction". Although the test technique essentially measures "insertion loss", it does so in a laboratory setting rather than the field environment to which insertion loss more generally refers. The term "transmission loss" is inappropriate because it refers to a different environment-corrected test result from the one used here.

By analyzing the test facility configuration, it can be shown that the measured value of noise reduction as defined above is approximately equivalent to a measured value of transmission loss. A comparison of measured values of noise reduction for a 2.54 mm (0.100 in) aluminum test panel (Panel J) with calculated values of transmission loss confirmed this equivalency: over the mass-law dominated region of from 500 to 4000 Hz, the noise reduction of continuous broadband noise measured in one-third octave bands differed from mass-law values by an average of less than 0.5 dB with a standard error of only 0.6 dB.

Noise reduction vs. frequency curves for the test panels are presented in Appendix B for both log frequency and linear frequency scales over the frequency range of from 100 to 2000 Hz. These curves are derived from the time delay spectrometry (TDS) measurement technique described earlier, and form the basis for interpreting the acoustical performance of each noise reduction concept as described in Section 5, in which superpositions of key noise reduction curves are presented to allow comparisons of panel performance. Also available for the acoustical interpretation described in Section 5 were some noise reduction vs. frequency curves obtained using continuous broadband noise and a one-third octave analysis.



- (1) Loudspeaker: Altec Lansing Type ER-15  
 (2) Microphone: Bruel and Kjaer Type 4145

Figure 20. Instrumentation Block Diagram.

## 5.0 ACOUSTICAL INTERPRETATION

Appendices A and B contain, respectively, a detailed description of the test panel configurations and the acoustic test results. This section presents an acoustical interpretation of the results.

### 5.1 Panels A and B

Without cavity absorption (configuration A.1) at frequencies less than  $f_0$ , there appears to be an increase in noise reduction (NR) above the mass-law value that can be explained by the reduction in cavity air stiffness due to the edge vents (see Figure 21). The NR curve shows a large dip over a frequency region above  $f_0$ , but then at higher frequencies adheres to the theoretical 18 dB per octave characteristics as though the cavity air stiffness is unchanged by the edge vents. This resumption occurs at approximately the frequency at which the period of vibration is equal to the time for a pressure wave to travel from the center of the panel to the edge (375 Hz for the 1.8 m (6.0 ft) panel dimension). Above this frequency it is reasonable to assume that the edge vents have no influence on the stiffness of air at the center of the panel.

With the addition of absorption in the cavity (configuration A.2), the NR values are reduced at frequencies less than  $f_0$  and increased above  $f_0$ . This occurs because the absorption material partially prevents the lateral movement of cavity air and hence no reduction in air stiffness results.

For Panel B with edge seals, the effect of adding cavity absorption is negligible at frequencies less than  $f_0$ , as expected (see Figure 22). At frequencies between  $f_0$  and 300 Hz, the absorption increases the NR by up to 6 dB. The curve for the cavity lattice lies in between these two cases and hence is only partially effective in reducing the build-up of lateral cavity modes.

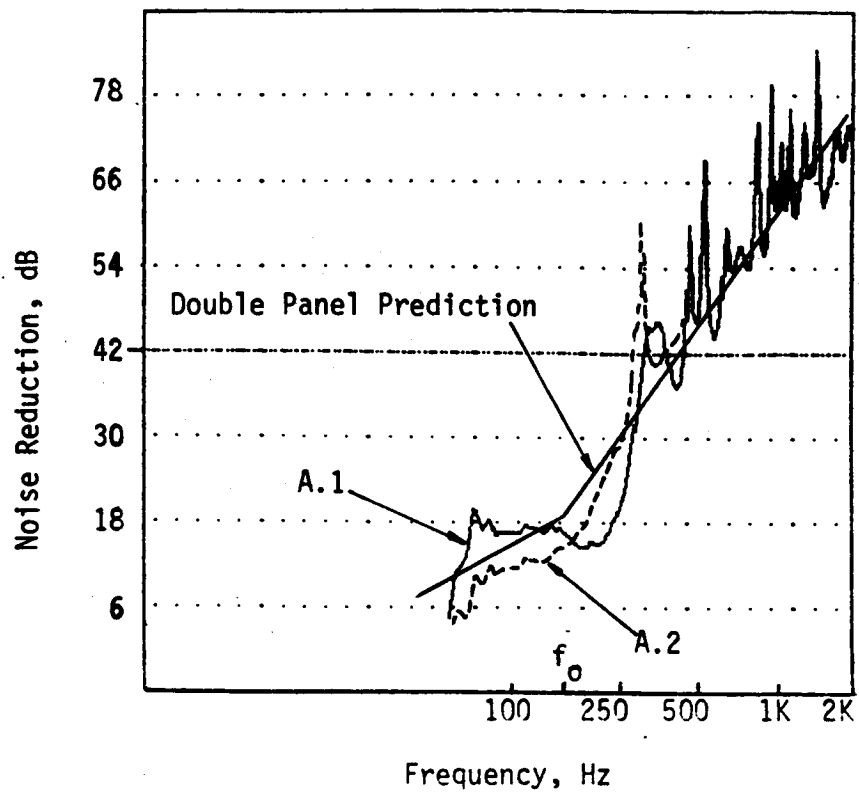


Figure 21. The Effect of Absorption on Double Panels Without Edge Seals. A.2 has absorption, A.1 has not.

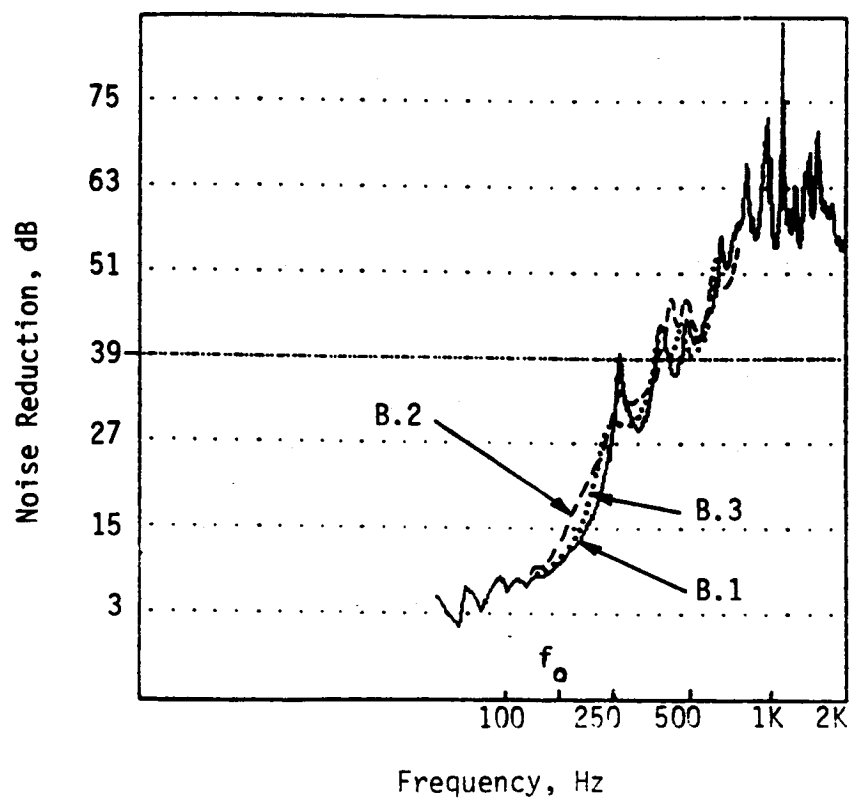


Figure 22. The Effect of Absorption and a Cavity Lattice on Double Panels with Edge Seals. B.1 has neither absorption nor a cavity lattice, B.2 has absorption only, and B.3 has a cavity lattice only.



Figures 23 and 24 show comparisons of the edge-seal and no-edge-seal panel designs without and with cavity absorption, respectively. Without absorption (Figure 23), there is a noticeable improvement of up to 12 dB at frequencies lower than  $f_0$ , but essentially little change at higher frequencies, except for isolated resonance, again illustrating the lowering of cavity stiffness. However, as noted above, the resonance at  $f_0$  still occurs for the no-edge-seal panel.

With absorption (Figure 24), there is an increase of about 5 dB at frequencies less than  $f_0$  for the no-edge-seal panel, together with a large increase near 300 Hz. The small bandwidth of this latter increase indicates that a resonant phenomenon is involved. The cause is not fully understood, but could be the result of a Helmholtz-type resonance. Additional measurements are required to optimize this effect. At higher frequencies, there appears to be a general increase in NR for the panel with no edge-seal, as verified by the one-third octave band data shown in Figure 25.

In conclusion, it appears that venting the cavity has the effect of increasing the NR of the double panels at frequencies less than  $f_0$  and at higher frequencies, but not at  $f_0$  itself. With cavity absorption, the increases are on the order of 5 dB for broadband noise. For pure tones, the increases at frequencies greater than  $f_0$  can be substantial at selected frequencies.

## 5.2 Panel C

Panel C was designed to test the concept of using Helmholtz resonators to increase the transmission loss of panels. Configuration C.2 was designed for a resonance at 155 Hz. The actual increase occurred slightly below 100 Hz, where an increase of about 4 to 5 dB is observed (see Figure 26). In all cases, the measured resonant frequency was lower than that predicted, indicating that the simplified expression used for calculating end effects of the resonator holes was unsatisfactory.

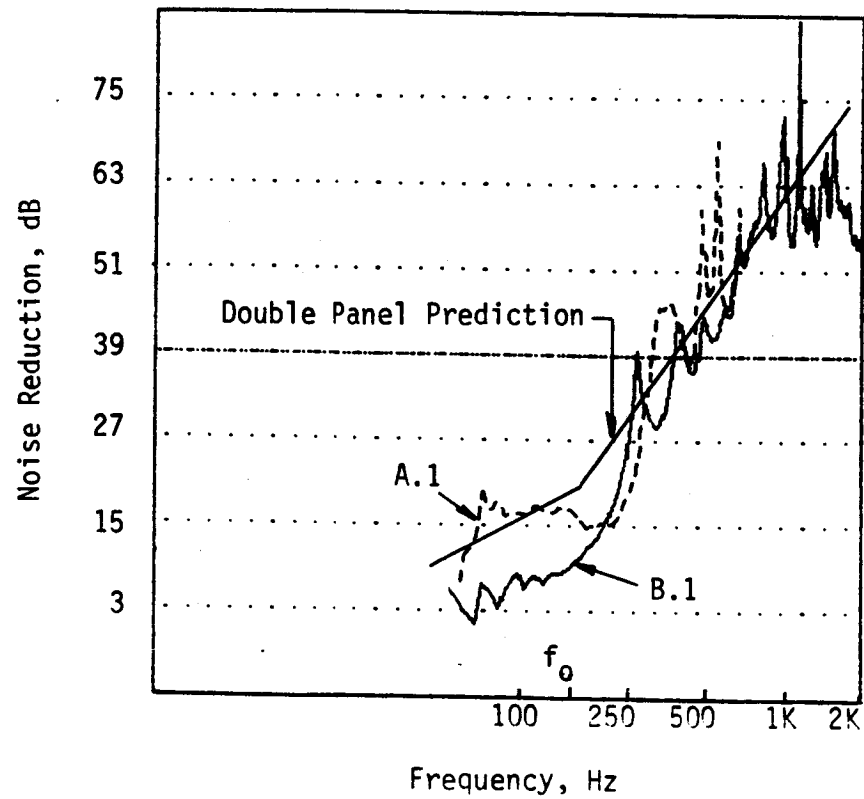


Figure 23. The Effect of Edge Seals on Double Panels Without Absorption. A.1 has no edge seals, B.1 has them.

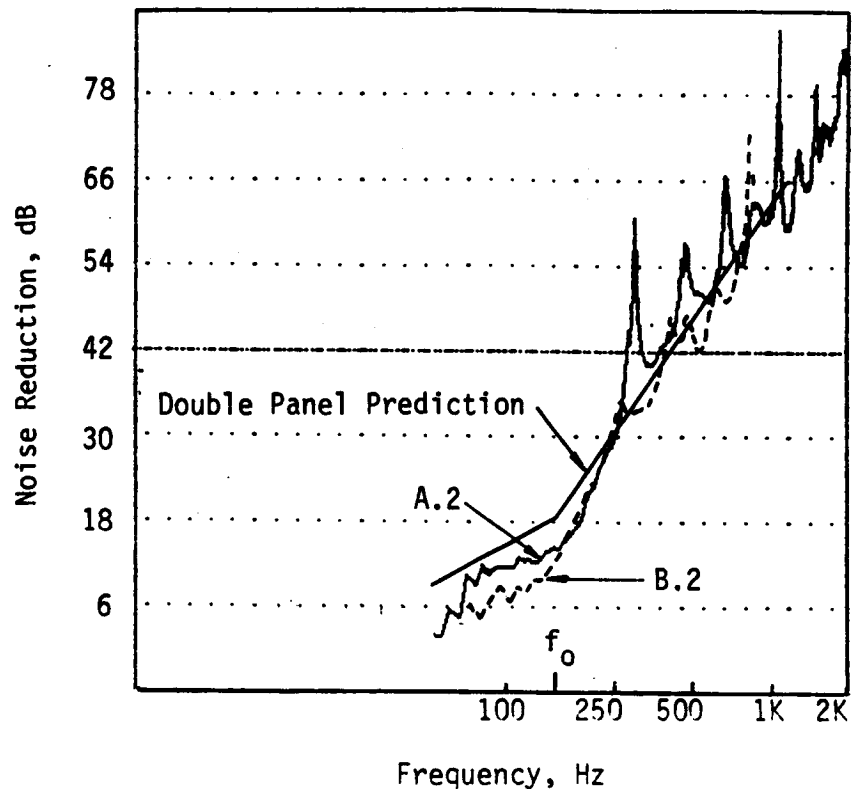


Figure 24. The Effect of Edge Seals on Double Panels With Absorption. A.2 has no edge seals, B.2 has them.

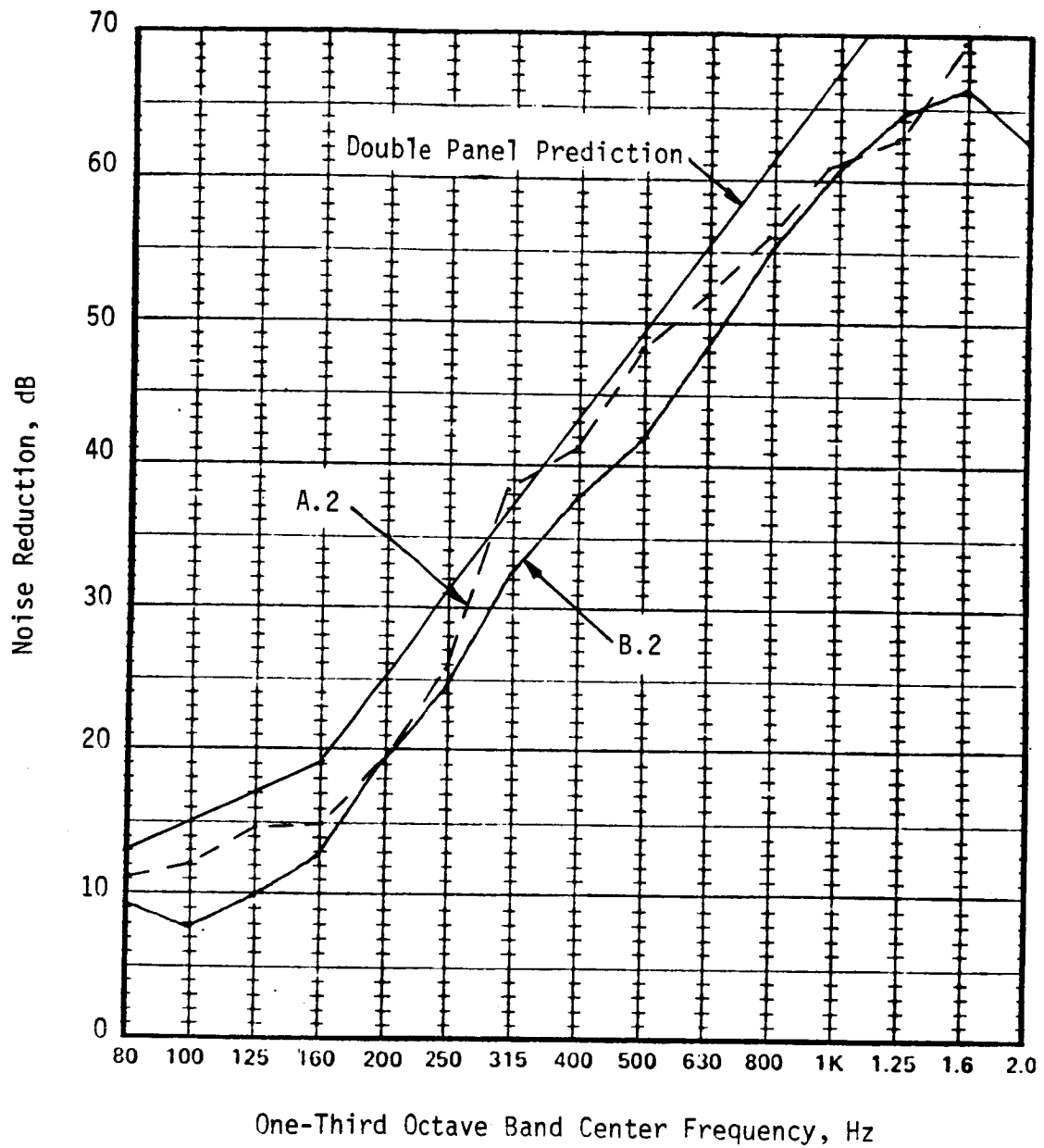


Figure 25. The Effect of Edge Seals on Double Panels With Absorption. A.2 has no edge seals, B.2 has them. From broadband noise test results.

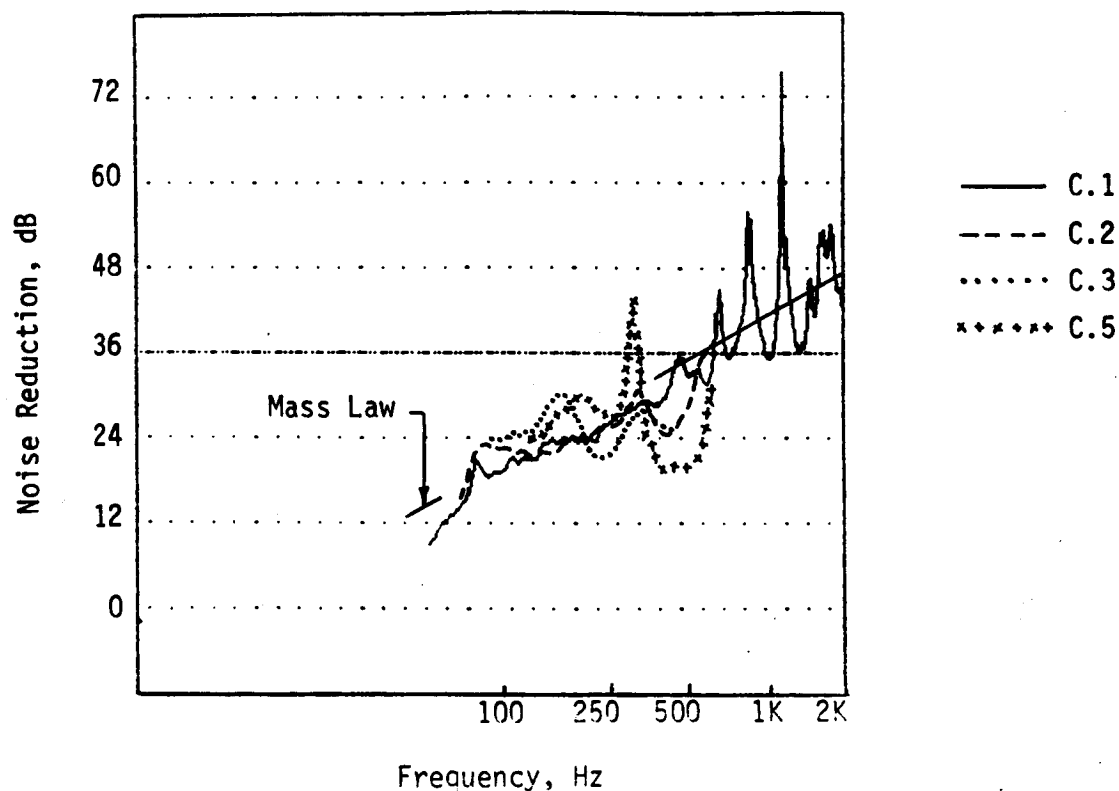


Figure 26. The Effect of Resonators on Single Panels. C.1 is the baseline case with no hole. Resonator hole diameter is smallest for C.2, largest for C.5. C.4 (not shown, see Appendix B) had tubes inserted in the holes and had a lower frequency noise reduction peak.

Increasing the diameter of the holes (the resonator necks - configuration C.3) increases the resonant frequency to slightly less than 200 Hz (the calculated frequency was 275 Hz). At this frequency, the increase in NR is on the order of 6 to 9 dB. In addition, there is an increase of about 5 dB at lower frequencies down to less than 100 Hz. However, in both configurations C.2 and C.3, the increases in NR are accompanied by rather noticeable decreases in certain higher frequency ranges. The cause of these dips in NR is not fully understood.

Inserting tubes in the holes to increase the effective mass of air in the resonator neck reduced the Helmholtz frequency as expected. Finally, removing the tubes and increasing once again the size of the holes produced a large increase in NR on the order of 12 dB at 300 Hz. Again, however, this was accompanied by a large reduction in NR at higher frequencies.

In conclusion, it appears that the inclusion of resonators in a panel can provide a significant increase in transmission loss over a narrow bandwidth - the higher values being obtained at higher frequencies with larger resonator holes, perhaps due to the increased radiation efficiency. Thus, to achieve large increases (i.e., 10 dB) in NR at lower frequencies (i.e., 160 Hz - the first turboprop harmonic), requires larger resonator holes than were used in these tests. Increases in NR are also observed at frequencies lower than the resonant frequency. In none of the test cases was there evidence of the double-panel fundamental resonance ( $f_0$ ). Further tests are required to investigate the reason for the large reductions in NR at higher frequencies, and to optimize resonator design including the desirability of adding absorption in the cavities.

### 5.3 Panel D

The NR for the baseline panel with cavity at ambient pressure agrees well with that predicted - see Figures 15 and B-11. In fact, a higher value of  $TL_M$  was measured than was calculated. As the cavity pressure was progressively reduced, the skins bowed slightly and compressed the resilient pads attached to the support blocks (see Figures A-7 and A-8). Perhaps for this reason the acoustic results were disappointing. There is perhaps slight

evidence of an increase in NR at frequencies just below  $f_0$ , as was predicted, but the results cannot be considered reliable. There was a substantial noise reduction loss evidenced between Panel D.1 (unpressurized) and Panel D.2 ( $-2.1 \times 10^4$  Pa or  $-3.0$  lb/in<sup>2</sup>), and this same loss persisted at higher decompressions. All theoretical evidence indicates that the predicted results should occur, but they were not demonstrated here.

#### 5.4 Panel E

The acoustical test results shown in Figures B-14 to B-20 show very little change in NR with the addition of damping material in either the tuned or the untuned condition. Without a damping effect, the results cannot be used to indicate whether the intrinsic tuning concept works or not. It would be worthwhile to re-test this panel at a later date using more damping material.

Impact response tests, however, showed that the center of the skin bays responded less when the panel was damped than when it was not. (The reduced response was independent of mass, since at each of three conditions both the damped and undamped panels were of equal mass - see Section 3.5.) Therefore, these tests provide a basis for determining whether the intrinsic tuning concept works or not by comparing the reduction in response using damping at two conditions: (a) the tuned condition, and (b) untuned conditions A and B described in Section 3.5.

The results are shown in Table 2.

TABLE 2. Impact Response of Panel E at Tuned and Untuned Conditions With and Without Damping

Condition	Panel Configurations Compared (damped vs undamped)	Reduction in Skin Bay Imaginary Response, percent
Untuned A	E.2 vs. E.3	31
Tuned	E.4 vs. E.5	26
Untuned B	E.6 vs. E.7	21

A significant reduction in response was exhibited through use of damping material - though perhaps not quite sufficient to translate to an acoustic benefit - but the response does not seem to be reduced more in the tuned condition than in the untuned ones.

#### 5.5 Panel F

Panel F was designed to have a fundamental panel resonance of 600 Hz when attached rigidly to the aircraft-type framing, thus providing high NR at low frequencies due to stiffness effects. This did not in fact materialize because the stiffness of the panel turned out to be comparable to that of the framework and so behaved almost independently of the framework. Moreover, the framework of the test panel was not firmly connected to the test opening and so did not provide the same stiffness that it would in a real aircraft application. Thus the NR of the double-panel construction was essentially that of two infinite panels - see Figure 27 - and the measured results agree very well with those predicted. The value of  $\Delta TL_M$  is 29 dB.

There is some indication of stiffness effects in the trend towards higher NR than calculated at 200 Hz (see also the pure-tone NR curve for F.1 in Figure 28) followed by what is perhaps a fundamental resonance mode at 250 Hz for a single panel formed between adjacent frames and longerons. The panel design and test in this case does not fully represent the concept of a stiff panel with high fundamental resonant frequency providing high NR at low frequencies. Further design is necessary to develop what is considered to be a promising concept.

#### 5.6 Panel G

Configuration G.1, which was designed to demonstrate the NR of the single resonator panel, showed that the fundamental double-panel frequency  $f_0$  is evident at 340 Hz with the 13 cm (5.3 in) square lattice. Note that this resonance was not evident with the smaller resonator lattice used in Panel C. This resonance reduces the NR of the complete construction of configuration G.2 in the frequency range 200 to 630 Hz, and eliminates the benefits of the Helmholtz resonance designed to occur at 200 Hz. This latter



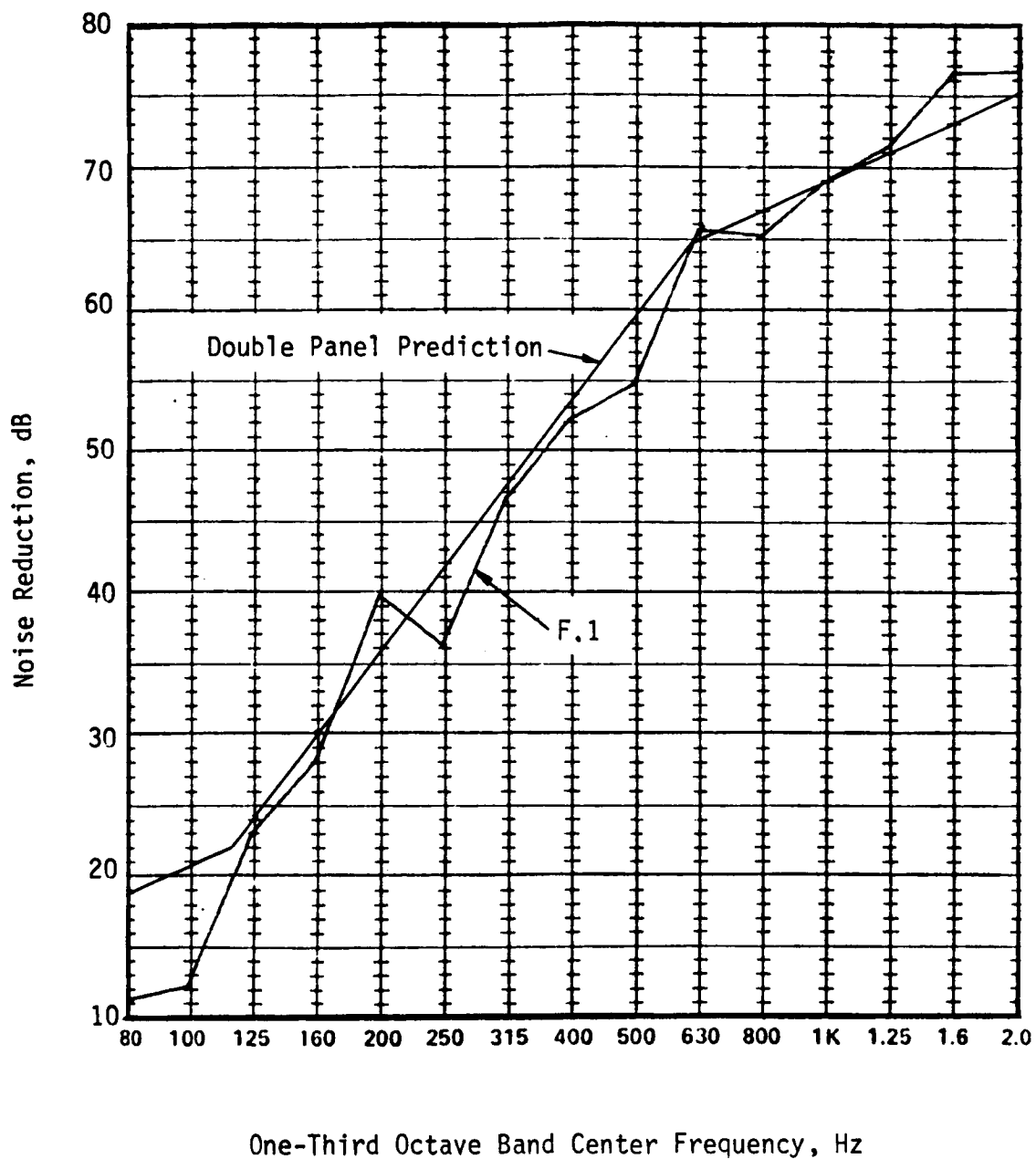


Figure 27. Noise Reduction of a Double Wall With One Stiff Panel and One Limp. From broadband noise test results.

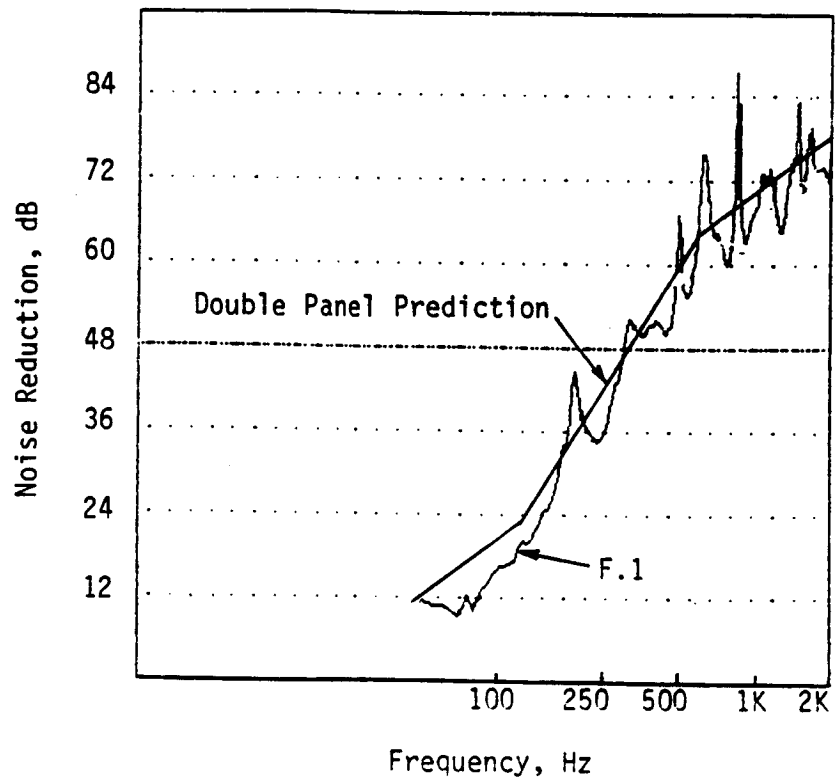


Figure 28. Noise Reduction of a Double Wall with One Stiff Panel and One Limp.

resonance is observed in the pure-tone NR curve, but is affected by other neighboring resonances. Otherwise, the measured result generally follows the prediction.

The results obtained from this construction indicated the resonator volume should be as small as possible to minimize the double-panel resonance ( $f_0$ ). As noted for Panel C, further work is needed to optimize the resonator design.

## 6.0 AIRCRAFT-APPLICATION DESIGNS

This section addresses the practicality of developing aircraft sidewalls which incorporate the acoustical concepts studied here. Practicality issues extend into several nonacoustical subject areas:

- (a) Structural integrity
- (b) Inner trim design - including aesthetics
- (c) Fire, smoke, heat release and toxicity
- (d) Thermal insulation
- (e) Moisture control
- (f) Producibility and maintainability
- (g) Weight

For each of these subject areas, design requirements were summarized and the various concepts evaluated against them. As a result of this evaluation practical designs were formulated of complete sidewalls (rather than the partial sidewall concepts of some of the test configurations). Acoustically-important parameters of each design were then adjusted to give a predicted noise attenuation which would result in 80 dB(A) within the passenger cabin of the study aircraft - see Section 1.0. The weight of the sidewalls was then calculated.

### 6.1 Nonacoustical Design Requirements

#### (a) Structural Integrity

The pressure shell and associated structure must be able to support cabin pressure and aircraft loads.

#### (b) Inner Trim Panel Design

The inner trim panel shall:

- o Protect acoustic and thermal hardware from damage

- o Resist impact from luggage/passengers below waistline height equivalent to a Gardner Impact of 1.6 N m (14 lb in)
- o Resist baggage and foot abrasion
- o Have facing materials which are easily cleaned and meet stain resistance requirements
- o Provide a uniform look down the length of the cabin both in terms of line and, if required, decor
- o Incorporate a window and accomodate a window shade installation
- o Incorporate services, such as air conditioning vents, as required
- o Accomodate tolerances of the fuselage structure, baggage racks and partitions without trimming
- o Be mounted to minimize the direct transmission of structureborne energy to the cabin.

(c) Fire, Smoke, Heat Release and Toxicity

The applicable fire protection requirements for materials used in passenger compartment sidewalls, such as interior wall panels, decorative trim, thermal and acoustic insulation and their coverings, etc., are defined in FAR Part 25 paragraph 25.853. This requirement specifies burn tests to demonstrate the self-extinguishing characteristics of the materials in terms of burn length and flame time following removal of the flame source as well as the flame time of drippings from the test material.

Smoke and heat release requirements are currently assessed by the NBS smoke test and the Ohio State University heat release test. The smoke test produces unitless numbers relating to the density of the smoke emitted from the test material when subjected to a heat

source. The test is in two parts, one with the gases emitted from the material under test ignited by a pilot light and one with the emitted gases not ignited. The heat release test again subjects the test material to a heat source and the additional heat output by the material measured. A visual inspection is also included to detect melting and the formation of char on the specimen. The formation of char while burning is acceptable but materials that drip or crumble while burning are unacceptable for use above the floor. The FAA is currently proposing to include smoke and heat release requirements in FAR Part 25 along with the permitted smoke and heat release levels. If such a regulation is adopted it would set restrictions on complete sidewall burn as well as on the constituent materials.

Toxicity is less well defined. The current toxic gas analysis relies on comparison of the toxic gas output of new materials with that of currently used materials.

(d) Thermal Insulation

The thermal design requirements are as follows:

- o The sidewall heat conductance from the cabin to the skin should ideally be less than  $0.68 \text{ W m}^{-2} (\text{°K})^{-1}$  ( $0.12 \text{ Btu h}^{-1} \text{ ft}^{-2} (\text{°F})^{-1}$ )
- o All structures connected to the outside skin such as support brackets, clips, miscellaneous attachments and handles should be of low thermal conductivity and isolated from the cabin wall
- o To prevent sustained vertical airflow into or from ceiling or underfloor areas, the airgaps between the cabin liner and fuselage outer skin should be blocked with airflow dams at floor and ceiling level
- o All fiberglass batting used for insulation blankets should be bagged.

(e) Moisture Control

The moisture control design requirements are as follows:

- o A continuous, low permeability, vapor barrier between the liner and skin is required. This vapor barrier should be incorporated into the inboard side of the sidewall insulation blankets and extend to seal around window torque boxes and doors
- o Sufficient drainage must be provided to prevent accumulation of moisture from frost and condensation on the inside surface of the fuselage outer skin and associated structure. Accumulated water causes corrosion and weight problems.

(f) Producibility and Maintainability

The sidewall design should:

- o Enable manufacturing costs to be kept to a minimum by using cost-effective manufacturing techniques
- o Employ readily available materials which meet the other design requirements
- o Enable fabrication and installation to be labor non-intensive.

The design should also permit:

- o Interior trim panels and assemblies within the sidewall to be readily removable and easily transported out of the aircraft
- o Access for window replacement
- o Access to inspect outer skin
- o Easy cleaning of inner trim panels and air grills.

(g) Weight

The weight of the sidewall should be the minimum to perform the required task. This requirement is, however, tempered by two further considerations, namely manufacturing costs and operational costs.

6.2 Nonacoustical Assessment (Except Weight)

Each of the seven acoustical concepts embodied in Panels A to G was examined for its ability to be incorporated into a sidewall design which meets the nonacoustical design requirements described above. (Weight was, however, considered later.) A summary of this assessment is presented in Table 3.

Having more features, these aircraft-application sidewalls are more complex to produce and are heavier than designs for turbofan aircraft which are required to attenuate mainly boundary layer noise. It was therefore assumed in the assessment that these designs would extend over a short length of fuselage, approximately 4.5 m (15 ft) in the case of the study aircraft. Because underfloor and ceiling regions of the fuselage generally have more space for conventional noise control treatments, the assessment focused on the sidewall region between the floor and the ceiling.

6.2.1 Panel A - Double Wall With Vented Cavity

(a) Structural Integrity

Venting into the lower sections of the fuselage is workable. Frame/longerons are required. If the inner panel is to be suspended by point connectors, severe fatigue problems could occur due to stress concentrations at connectors from the transfer of shearing forces.

(b) Inner Trim Design

Can readily meet requirements.



TABLE 3 - NONACOUSTICAL CHARACTERISTICS OF AIRCRAFT-APPLICATION SIDEWALL DESIGNS

Design Principle	Related Test Panel	Sidewall Design in Figure	(a) Structural Integrity	(b) Inner Trim Design	(c) Fire Smoke Toxicity	(d) Thermal	(e) Moisture	(f) Producibility and Maintainability
Double Wall with Vented Cavity	A	30	Workable	O.K.	O.K.	O.K.	O.K.	O.K.
Double Wall with Cavity Lattice	B	31	Workable if not bonded to outer skin	O.K.	O.K.	O.K.	Workable	Problems
Wall Resonators	C	32/33	Workable if not bonded to outer skin	O.K.	O.K.	Resonator to be made of low conductivity material	Resonators must be drained	Problems
Double Wall with Reduced Cavity Pressure	D	34	Major problems	Possible aesthetic problems	O.K.	Workable	O.K.	Outer skin must be removable for inspection
Skin-Longeron Tuning	E	35	O.K.	O.K.	O.K.	O.K.	O.K.	O.K.
Double Wall with Stiff Panel	F	36	Workable. Extensive analyses and testing req'd	Workable with added inner trim panel	O.K.	O.K.	O.K.	O.K. with added inner trim panel
Double Wall with Resonators	G	37/38	Workable. Extensive analyses and testing req'd	Workable with added inner trim panel	O.K.	Resonator to be made of low conductivity material	Resonators must be drained	Problems

(c) Fire, Smoke, Heat Release and Toxicity

Suitable materials are available.

(d) Thermal Insulation

A 5 cm (2 in) thick insulation blanket is required. A low emissivity reflective coating is required on the side of the blanket nearest the aircraft skin. Point connectors should be of a low thermal conductivity, non-metallic, material.

(e) Moisture Control

Flexible airflow dams (diaphragms which will permit oscillations of the air but prevent air transfer) between skin and liner are required at floor and ceiling levels. Pressure oscillations are acceptable, but sustained vertical airflow must be controlled. Lateral venting must be limited to minimize molecular water vapor flow. Moisture drains through airflow dams will be required. A continuous vapor barrier should be installed on the inboard cover of the insulation blanket.

(f) Producibility and Maintainability

Can readily meet requirements.

6.2.2 Panel B - Double Wall With Cavity Lattice

(a) Structural Integrity

Extensive stress and fatigue analyses are required if the lattice is to be attached to the outer, load bearing skin as a replacement for frames and longerons. If the lattice is to be attached to frames and window torque box it should be through flexible connectors so as not to carry structural loads - no unusual stress analysis required.

(b) Inner Trim Design

Can readily meet requirements.

(c) Fire, Smoke, Heat Release and Toxicity

Suitable materials are available.

(d) Thermal Insulation

A 5 cm (2 in) thick insulation blanket is required. The blanket could either be installed within each individual lattice bay, inboard of the lattice, or in two layers on opposing sides of the lattice. A low emissivity reflective coating is required on the outboard side of the insulation blanket(s).

(e) Moisture Control

Airflow dams, either solid or flexible, are required. A continuous vapor barrier is required on the inboard insulation cover. This could be difficult if insulation is only installed within the lattice bays. Design appears to allow for normal vertical drainage. Drainage through the airflow dams is required. The lattice should be moisture resistant.

(f) Producibility and Maintainability

Lattice requires extensive cut-outs to clear longerons and the window shade installation. Sculpturing of the lattice is required to accomodate a contoured inner trim panel. Manufacture, installation and maintenance cost of the lattice could be excessive. Manufacturing and installation time would be high if individual insulation blankets are required within each lattice bay. In addition it may be difficult to ensure a total coverage of the sidewall using this technique.

### 6.2.3 Panel C - Wall Resonator

#### (a) Structural Integrity

Frame/longerons could be eliminated if resonators are bonded to the outer skin but extensive analysis and structural testing would be required. This method is not recommended. It is suggested that the resonators be designed not to carry structural loads.

#### (b) Inner Trim Design

A method to mount the inner trim panel is required.

#### (c) Fire, Smoke, Heat Release and Toxicity

Suitable materials are available.

#### (d) Thermal (Insulation)

A 2.5 cm (1 in) layer of insulation should be added inboard of the resonators. Outer surface of inboard resonator cover and the outer surface of the added insulation blanket should have a low emissivity coating. Resonators should be of low conductivity material.

#### (e) Moisture Control

Airflow dams, either solid or flexible, are required. Resonators and airflow dams must contain drains. A continuous vapor barrier should be built into the inboard cover of the added insulation blanket.

#### (f) Producibility and Maintainability

Manufacturing problems are foreseen - particularly if extended necks are required for the resonators. Inspection of the resonator connection with the skin could be difficult.

#### 6.2.4 Panel D - Double Wall With Reduced Cavity Pressure

The aircraft-application sidewall assessed here is one in which the space between the inner and outer sidewall panels is vented to ambient.

##### (a) Structural Integrity

Major problems are seen in connecting the inner, pressurized skin and the outer skin. Skin stressing problems may exist due to fuselage bending. Structure, including inner and outer skins, would need analysis by finite element methods. Entire sidewall assembly would have to be designed to be fail-safe. Fatigue and fracture analysis are required. Transition between this sidewall and the conventional sidewall may create problems.

##### (b) Inner Trim Design

Inner trim panel would have to be installed inboard of the inner pressurized skin. Contouring of the inner trim panel around the windows may be impossible due to the presence of the pressurized skin. This could significantly affect the internal aesthetics. However, reducing the frame depth would enable a contoured inner trim panel to be used.

##### (c) Fire, Smoke, Heat Release and Toxicity

Suitable materials are available.

##### (d) Thermal Insulation

A 2.5 cm (1 in) thick insulation blanket should be attached to the outboard side of the inner skin allowing the outboard area to breathe. The convection thermal resistance of the air gaps will be more than doubled by the low density ambient air at cruise altitudes. The insulation blanket should have a low emissivity coating on its external cover. This will essentially eliminate

the predominant radiation heat transfer mode. Frames or jambs connecting the external skin to the offset pressurized inner skin should be of low thermal conductivity material such as titanium or stainless steel. A 0.6 cm (0.25 inch) non-metallic honeycomb inner liner inside the pressure skin is recommended to eliminate a possible cold wall.

(e) Moisture Control

Airflow dams should not be required. A vapor barrier may not be required with the incorporation of the insulation blanket mentioned above. Venting should be to the outboard, low vapor pressure side of the cavity. The low ambient pressure will greatly reduce any water vapor which might collect inside the insulation blanket. Any residual moisture will tend to migrate from the insulation and collect as frost on the inner surface of the cold external skin. Over a period of time this frost will sublime and migrate by molecular diffusion to the extremely low vapor pressure area that, at cruise altitudes, exists outside the external skin. Water drains would be required at the bottom of the external skin. These should ideally be combined with the atmospheric pressure equalization vents.

(f) Producibility and Maintainability

The complete outer, unpressurized skin must be removable to permit inspection. Localized design changes will be required to keep services such as air conditioning ducts out of this part of the fuselage.

6.2.5 Panel E - Skin-Longeron Tuning

(a) Structural Integrity

No unusual structural analyses are required.

(b) Inner Trim Design

Conventional designs could be applied.

(c) Fire, Smoke, Heat Release and Toxicity

Suitable materials are available.

(d) Thermal Insulation

A two inch thick insulation blanket is required. The outer surface of the insulation blanket should have a low emissivity coating.

(e) Moisture Control

Airflow dams, either solid or flexible, are required. A continuous vapor barrier should be incorporated into the inboard cover of the insulation blanket. Drainage through the airflow dams is required.

(f) Producibility and Maintainability

No problems are envisaged.

6.2.6 Panel F - Double Wall With Stiff Panel

(a) Structural Integrity

Honeycomb skin requires extensive analyses and structural testing. May require bonding of frames/longerons to honeycomb panel. Fewer frames and longerons might be needed. Large honeycomb skin sections could be inserted within a wide frame/longeron grid. Outside skin of honeycomb used over the 4.5 m (15 ft) length must conform with the skin line of the rest of the fuselage.

(b) Inner Trim Design

A more suitable inner trim panel is required. This should be sculptured to maintain the overall uniformity of the cabin sidewall appearance.

(c) Fire, Smoke, Heat Release and Toxicity.

Suitable materials are available.

(d) Thermal Insulation

Honeycomb has a higher thermal resistance than aluminum and hence the amount of insulation required may be reduced. A minimum of 3.8 cm (1.5 in) of insulation appears to be necessary. A low emissivity coating is required on the outboard side of the insulation blanket.

(e) Moisture Control

Airflow dams, solid or flexible, are required. A continuous vapor barrier is required on the inboard side of the insulation blanket. Drains in the airflow dams are required.

(f) Producibility and Maintainability

Can meet requirements with a suitable inner trim panel.

6.2.7 Panel G - Double Wall With Resonators

(a) Structural Integrity

Frame/longerons can be eliminated if resonators were bonded to outer skin. Extensive analysis and structural testing are required. This method is not recommended. Resonators should be designed not to carry any structural loads.



(b) Inner Trim Design

A more suitable inner trim panel, sculptured for aesthetic reasons, is required.

(c) Fire, Smoke, Heat Release and Toxicity

Suitable materials are available.

(d) Thermal Insulation

A 2.5 cm (1 in) thick insulation blanket should be installed inboard of the resonators. The outer surface of the inboard resonator cover and the outer surface of the added insulation blanket should have a low emissivity coating. Resonators should be of a low conductivity material.

(e) Moisture Control

Vertical airflow dams, either solid or flexible, are required. A continuous vapor barrier should be incorporated into the inboard cover of the insulation blanket. Resonators and airflow dams must contain drains.

(f) Producibility and Maintainability

Manufacturing problems are foreseen. Inspection of the resonator connection with the skin could be difficult. Can meet other requirements with a suitable inner trim panel.

### 6.3 Design Descriptions

Aircraft-application sidewall designs which emerged from the earlier assessment of the ability of the acoustical concepts to meet aircraft design requirements are described below. These designs were first formulated before the acoustical test results were available, and were adjusted after the

acoustical assessment described in the next section was performed. The designs are schematic and require considerable further analysis and testing to develop.

Figure 29 describes a conventional aircraft sidewall design to permit comparison with the new designs illustrated in Figures 30 to 38.

#### 6.3.1 Panel A - Double Wall With Vented Cavity

The design (Figure 30) is the same as a conventional sidewall with the exception of the airflow dams. The conventional airflow dams are replaced by a membrane-like diaphragm (not shown). This diaphragm is sufficiently light and flexible to permit relatively unrestricted air oscillation due to acoustic excitation while preventing mean, i.e. non-oscillatory, airflow between the sidewall and the ceiling or underfloor areas. Attention may need to be given to preventing noise transmission from these areas to the cabin.

#### 6.3.2 Panel B - Double Wall With Cavity Lattice

To prevent manufacturing problems the lattice passes over the longerons. The insulation blanket is continuous and installed either side of the lattice, thus enhancing the thermal protection. The design is described in Figure 31.

#### 6.3.3 Panel C - Wall Resonator

Two options for the installation of resonators are proposed.

Option A (Figure 32) is a design in which resonators are inserted between the longerons. The resonator cell walls are flexible where they meet the outer skin to ensure a good, corrosion-free, seal. The lower part of the resonator cells is designed to drain any collected moisture through the resonator hole.

Option B (Figure 33) has more resonators than Option A, and an easier installation method. (Sealing of the resonator cells is performed as part of

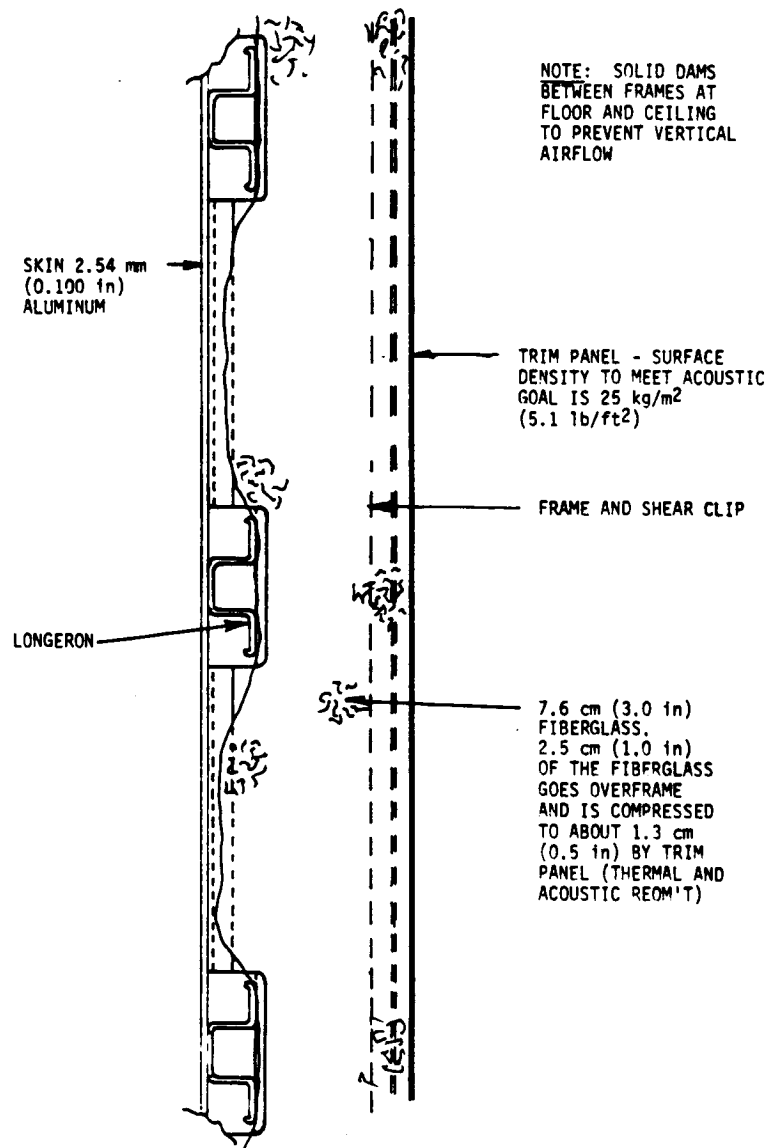


Figure 29. Conventional Sidewall Designed to Meet Turboprop Acoustic Requirement Using Typical Turbofan Aircraft Technology

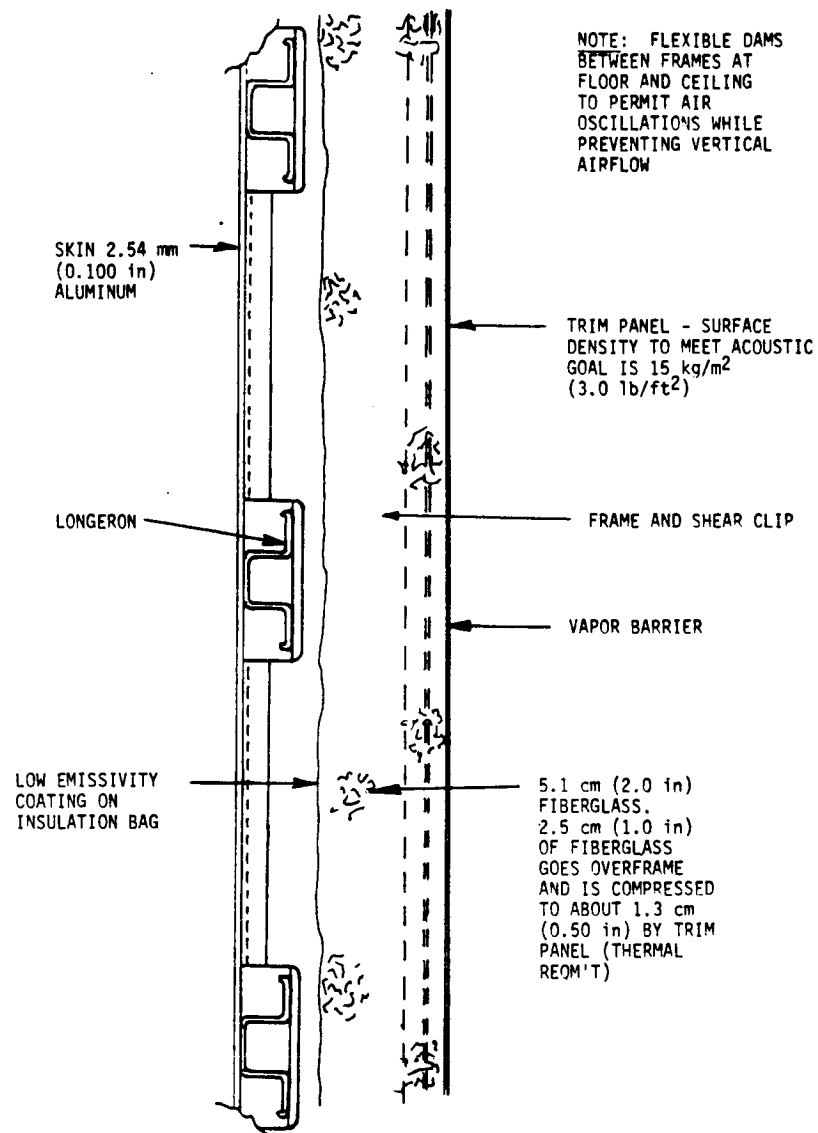


Figure 30. Sidewall Design to Permit Cavity Venting

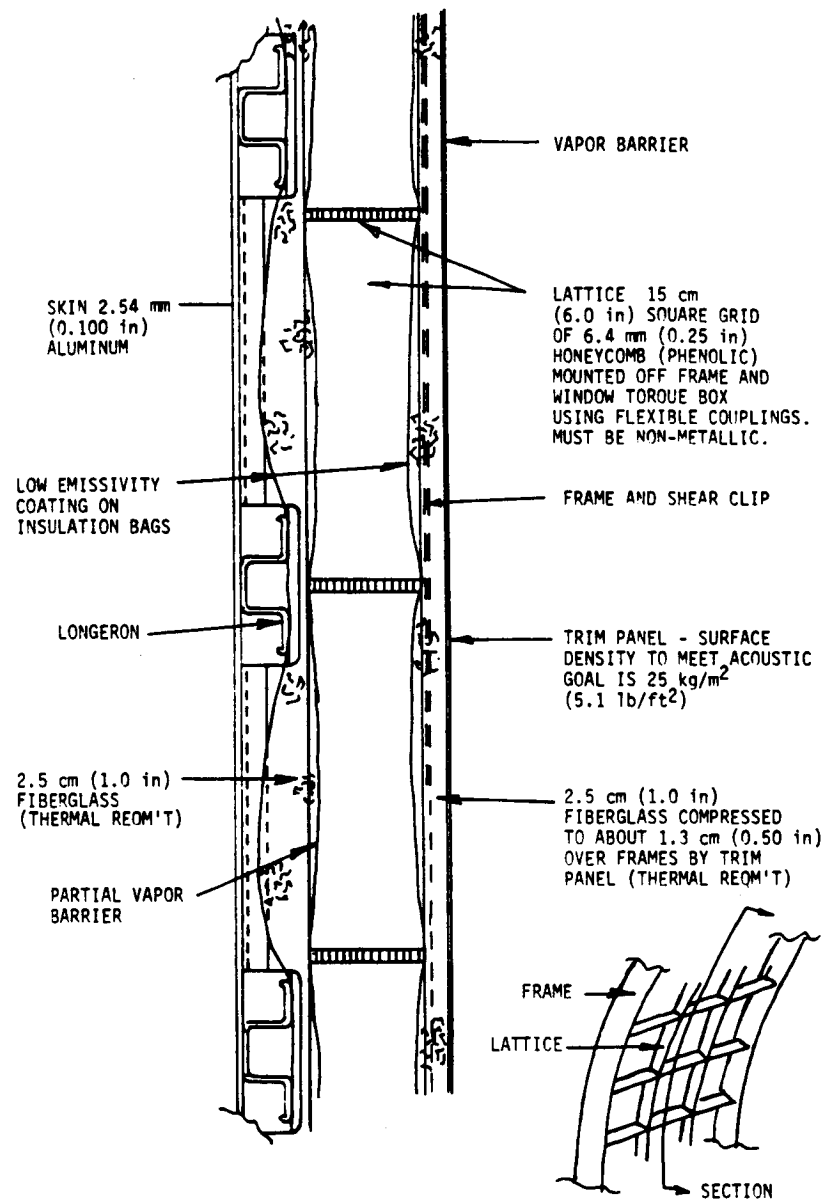


Figure 31. Sidewall Design with Cavity Lattice

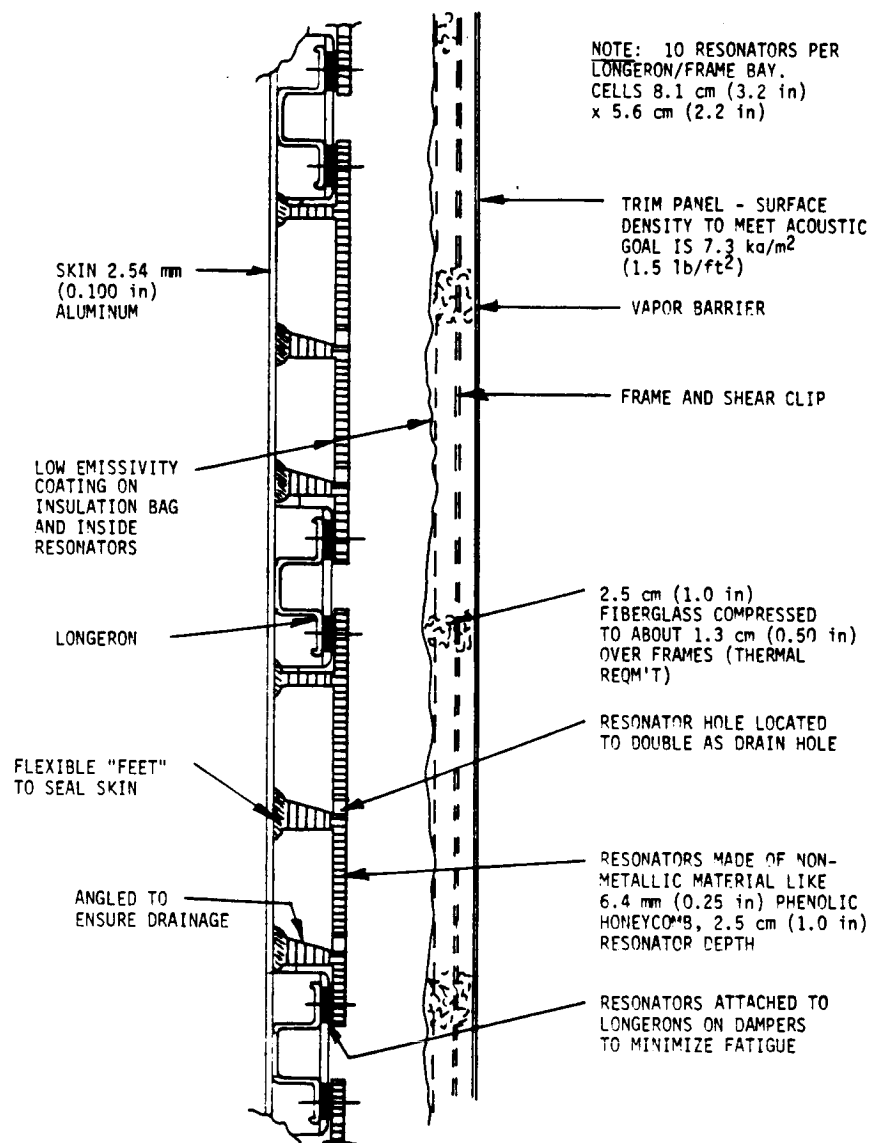


Figure 32. Sidewall Design with Wall Resonators Using Conventional Trim Panel (Option A)

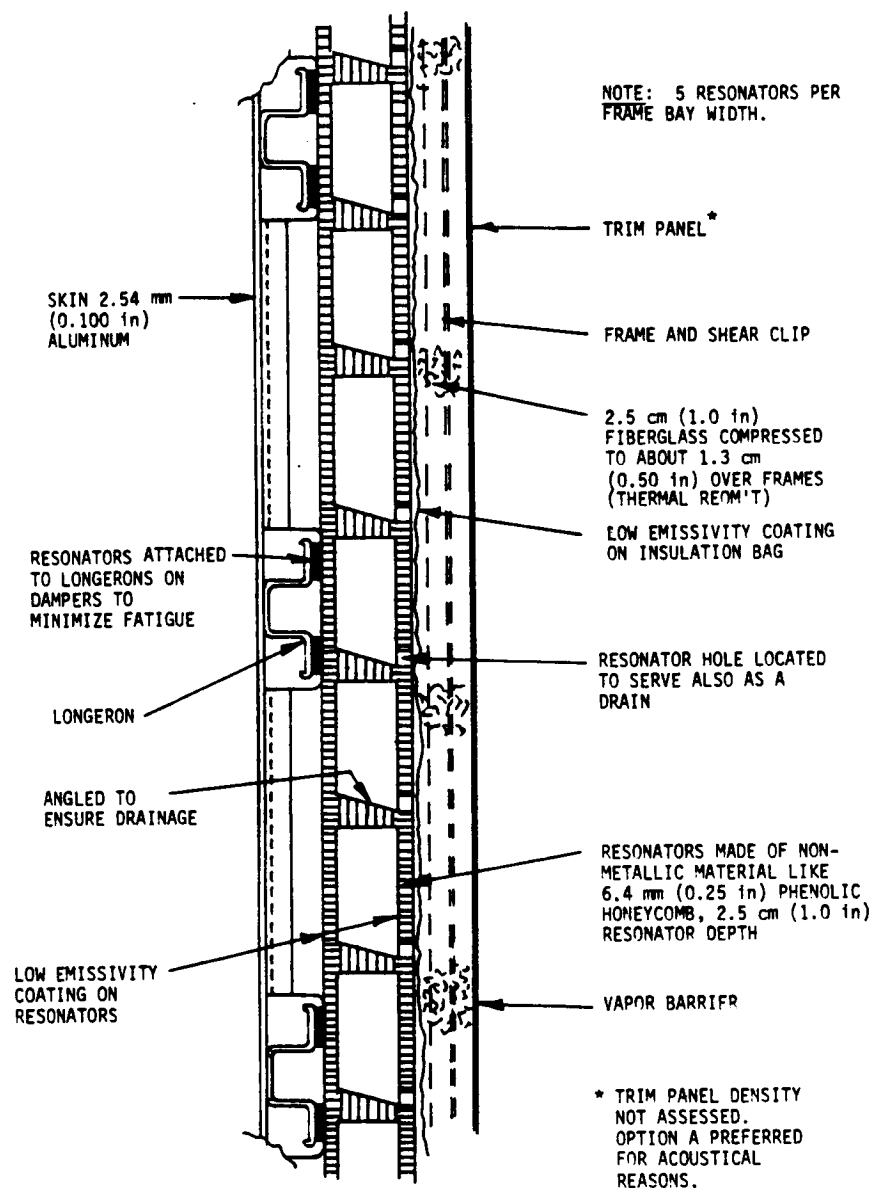


Figure 33. Sidewall Design with Wall Resonators Using Conventional Trim Panel (Option B)

resonator manufacture rather than during installation.)

#### 6.3.4 Panel D - Double Wall With Reduced Cavity Pressure

The design is explained in Figure 34.

#### 6.3.5 Panel E - Skin-Longeron Tuning

This design is similar to a conventional design except for longeron spacing and the application of damping material (see Figure 35).

#### 6.3.6 Panel F - Double Wall With Stiff Panel

The design (Figure 36) has two alternatives for the outer honeycomb wall, one where the honeycomb panel is an integral, load-carrying structure and an alternative where individual honeycomb panels are mounted in a load-carrying grid - much as some access panels are currently installed. It is expected that the acoustically-required degree of limpness can be provided in a single, i.e., homogeneous, trim panel that is nonacoustically satisfactory; however a construction concept has not yet been defined.

#### 6.3.7 Panel G - Double Wall With Resonators

This design (Figures 37 and 38) is a combination of the resonator design of Panel C and an acoustically-limp trim panel yet to be defined.

### 6.4 Acoustical Assessment

The aircraft-application designs described in the previous section were analyzed acoustically for the purpose of defining the trim panel mass necessary to just meet the acoustical design requirement. These values are given in Figures 29 to 38. It should be noted that trim panel mass is only one of a number of acoustically-important parameters that could be adjusted, and that this analysis did not therefore establish an optimum sidewall



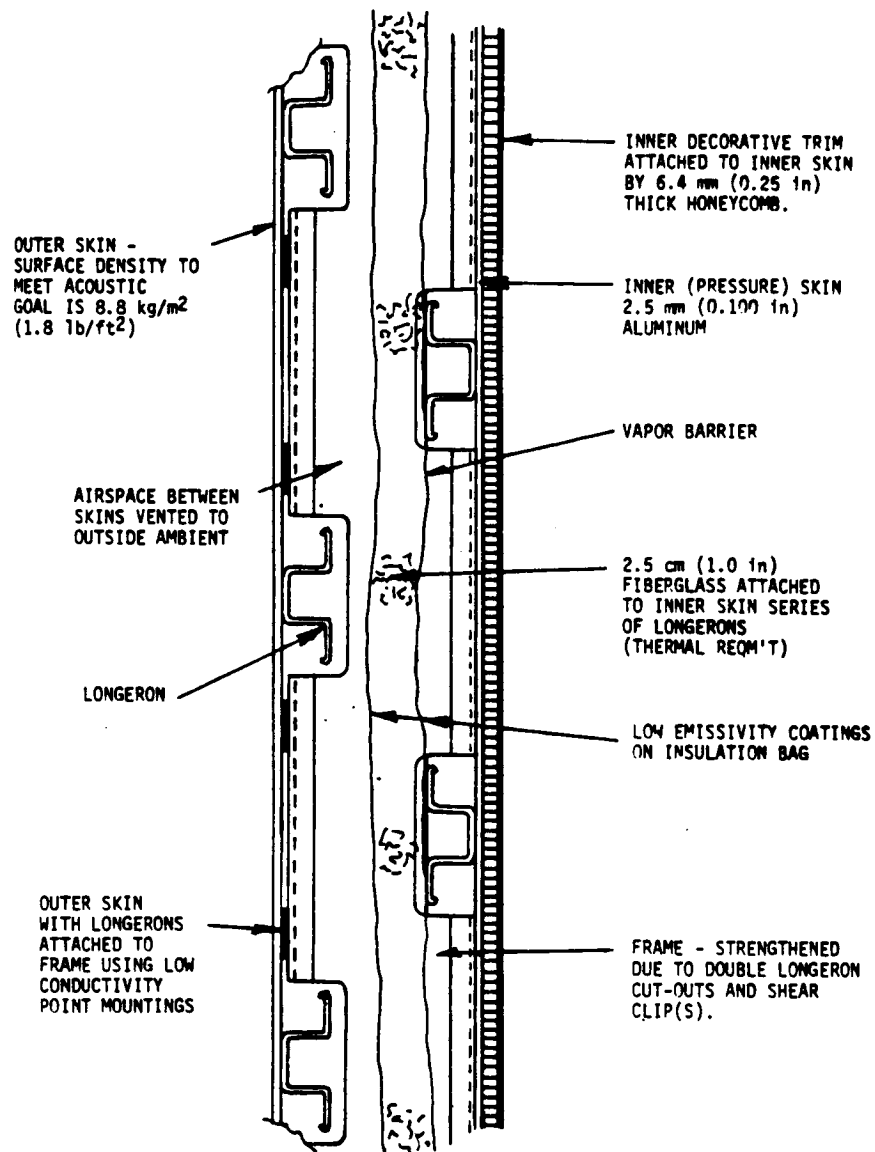


Figure 34. Double Wall with Reduced Cavity Pressure

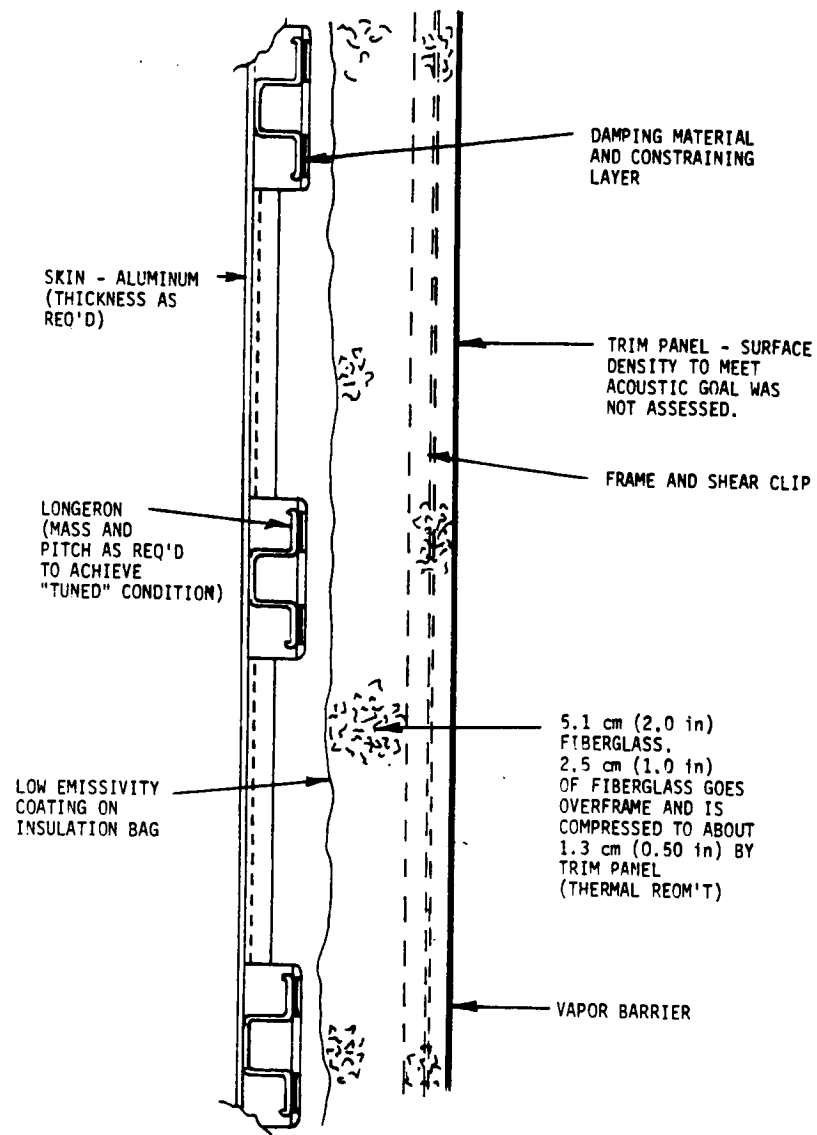


Figure 35. Sidewall Design for Skin-Longeron Tuning

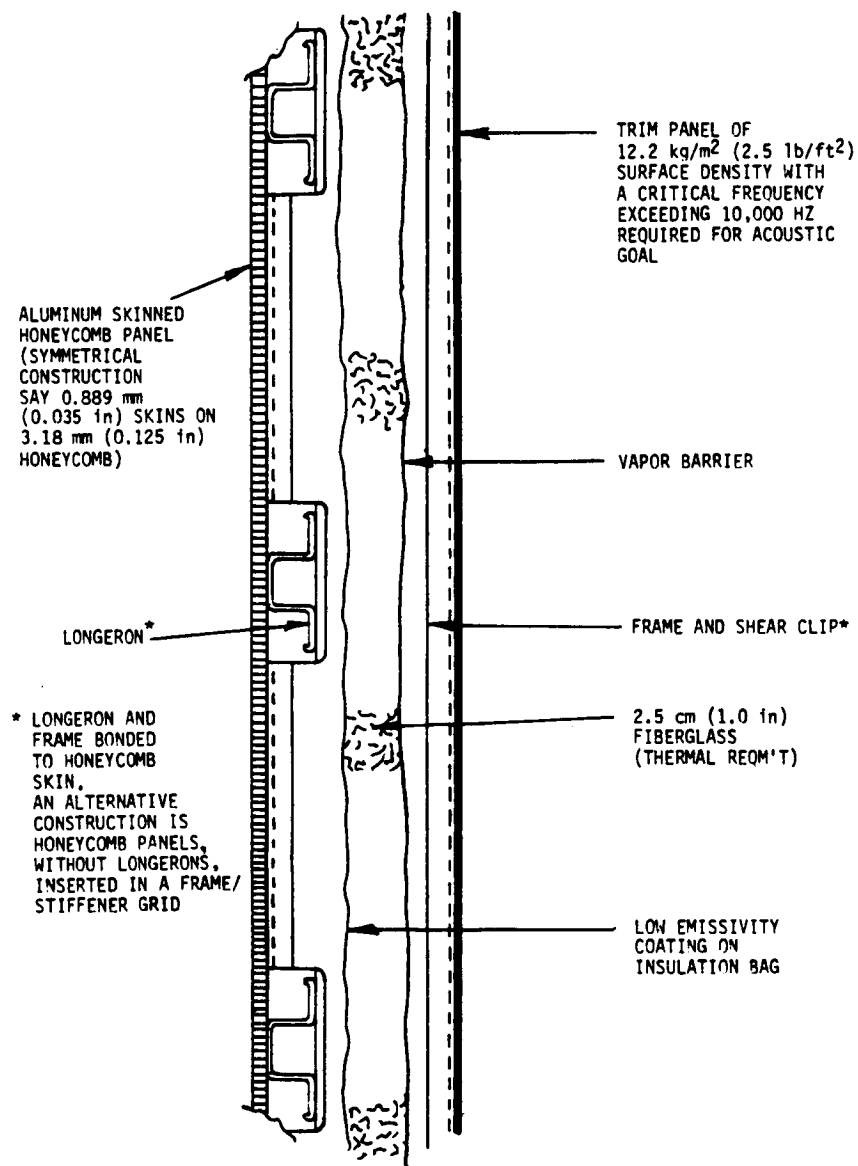


Figure 36. Sidewall with Stiff Outer Panel and Limp Trim Panel

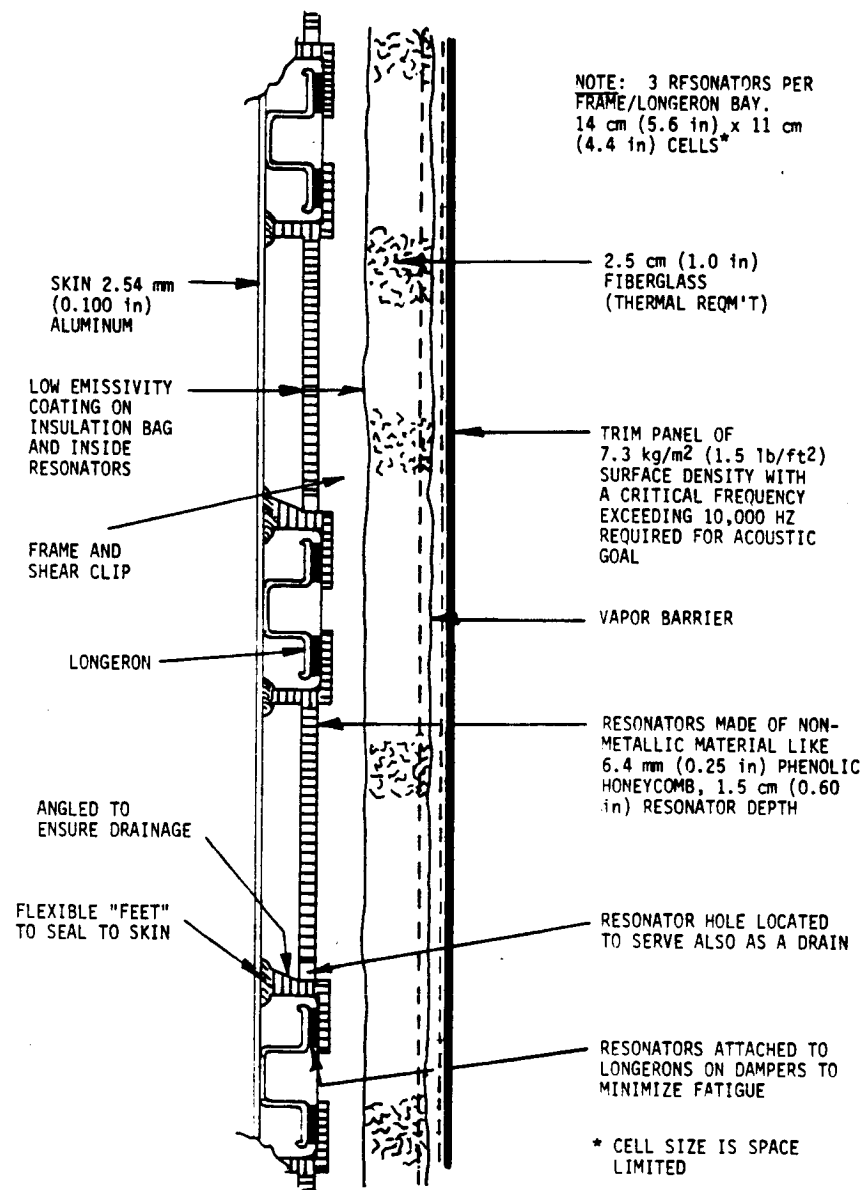


Figure 37. Sidewall Design with Resonators and Limp Trim Panel (Option A)

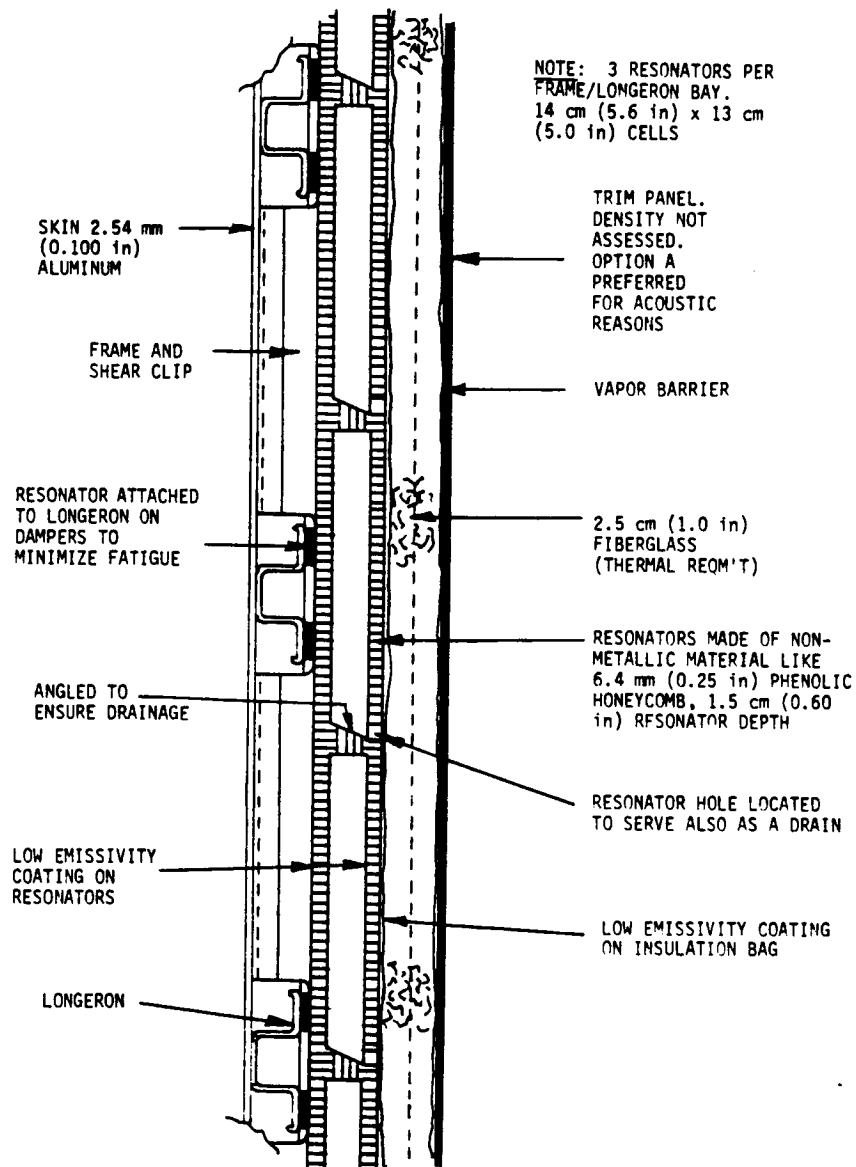


Figure 38. Sidewall Design with Resonators and Limp Trim Panel (Option B)



design. It did, however, go some way to permitting the various sidewall weights to be compared on the basis of equal acoustical performance. The weight analysis is described in the next section.

It should be noted that the designs were formulated to meet the acoustical requirement at the location of the highest exterior sound level, and that considerable relaxation of the acoustical requirements is possible outside this region. This might take the form of reduced trim panel mass. It should also be noted that no analysis has yet been performed of the ability of these designs to reduce other aircraft noise sources such as boundary layer noise. However satisfactory performance is likely.

For the purpose of the acoustical analysis, the fiberglass batting was specified to have a density of  $16 \text{ kg/m}^3$  ( $1.0 \text{ lb/ft}^3$ ).

For Panel C, Option A (Figure 32) is preferred over Option B (Figure 33) because there is a greater air volume immediately outside the resonator neck. The same conclusion applies to Panel G. Only Options A were considered further.

The analysis utilized the results of the panel tests, but assumed that resonators could be developed in further laboratory testing to perform rather better than the primitive versions tested here. The analysis did not, however, assume successful functioning of concepts that were tested and found not to work, such as Panel D's depressurized cavity. For these concepts, the aircraft-application designs presented earlier, and the weight assessment described next, should be regarded as reference material which may be useful if the acoustical principles are proven in further work.

## 6.5 Weight Assessment

The estimated weights of the aircraft-application sidewall designs are shown in Table 4. Also included for reference is the weight of a conventional turbofan aircraft sidewall designed to meet a similar design goal and estimated in the same manner. The reference panel weight used proven estimates

TABLE 4. WEIGHTS AND MERITS OF AIRCRAFT-APPLICATION SIDEWALL DESIGNS

Design Principle	Related Test Panel	Aircraft-Application Design Description	Sidewall Design in Figure	Sidewall Surface Density kg/m <sup>2</sup> lb/ft <sup>2</sup>	Comments
Conventional TF	-	Conventional Skin and Trim Panel	-	12 2.5	Turbofan (TF) design which does not meet turboprop requirement
Conventional TP	-	Conventional Skin and Trim Panel	29	42 8.5	Turbofan design adjusted to meet turboprop (TP) requirement
Double Wall with Vented Cavity	A	Conventional Skin and Trim Panel, Cavity Vented to Underfloor and Ceiling	30	32 6.5	Note (1)
Double Wall with Cavity Lattice	B	Conventional Skin and Trim Panel, with Cavity Lattice	31	44 9.0	Note (2)
Wall Resonators	C	Resonators Attached Inside Conventional Skin, with Conventional Trim Panel	32	32 6.5	Note (1)
Double Wall with Reduced Cavity Pressure	D	Conventional Skin, Unpressurized Cavity, Aluminum Inner Skin Bonded to Trim Panel with Honeycomb Sandwich	34	37 7.5	Notes (3) and (4)
Skin-Longeron Tuning	E	Conventional Skin and Trim Panel, Longeron Spacing Adjusted for Tuned Skin Bay/Longeron Resonance at Dominant Passenger-Response Frequency, Longérons Damped	35	- -	Note (3)
Double Wall with Stiff Panel	F	Aluminum-Honeycomb Sandwich Skin, with Limp Trim Panel	36	27 5.5	Note (1)
Double Wall with Resonators	G	Resonators Attached Inside Conventional Skin, with Limp Trim Panel	37	29 6.0	Note (1)

- (1) Recommended for further investigation  
 (2) Not recommended for further investigation  
 (3) Not recommended for further investigation without clearer evidence of functioning of acoustic principle  
 (4) The weight of this sidewall was calculated without assumption of an acoustical benefit from the depressurized cavity

of average sidewall weight including skin and stiffeners, inner trim panel, insulation and attachments, for a sidewall without windows. For the other panels, the reference panel weight was adjusted to reflect the increased skin weight required to meet structural integrity requirements, the changes within the sidewall and the revised trim panel required to achieve the interior noise level goal of 80 dB(A).

The weights given in Table 4 are for sidewalls without windows, which were not addressed in the present study, and refer only to the region of greatest exterior sound level. They are for comparison purposes only; further work is needed before they can be used for aircraft design purposes.

It can be seen in Table 4 that the weight of a conventional sidewall designed for the turboprop environment is very much higher than one designed for a turbofan environment, but that some of the designs studied here have weights which exceed the turbofan aircraft sidewall weight by a much reduced margin.



## 7.0 CONCLUSIONS AND RECOMMENDATIONS

Seven acoustical concepts were analyzed, and about thirty sidewall panel configurations embodying these concepts were tested for transmission loss, with the objective of reducing sidewall weight on advanced turboprop aircraft.

The following were recommended for further investigation as potentially-practical sidewall concepts offering weight savings over conventional designs. Investigation of combinations of these concepts is also recommended.

- o Design to Permit Cavity Venting. This sidewall is vented to the aircraft underfloor region and to the ceiling areas. Suitable diaphragms are included at floor and ceiling to prevent a mean flow of air across these boundaries.
- o Double Wall with Stiff Panel. This sidewall has a stiff outer wall probably composed of aluminum-skinned honeycomb, and a limp inner wall (trim panel).
- o Design with Wall Resonators and Limp Trim Panel. This sidewall has Helmholtz resonators attached to the skin inside the cavity, and a limp trim panel.
- o Design with Wall Resonators and Conventional Trim Panel. As above, plus use of a conventional-type trim panel.

The following was not recommended for further investigation:

- o Design with Cavity Lattice. This concept works acoustically but does not appear superior to use of cavity absorptive material, which is needed for thermal reasons.

The following were not recommended for further investigation without clearer evidence of functioning of the acoustic principles involved:

- o Double Wall with Reduced Cavity Pressure. This concept did not work acoustically in the present study. If proven to function acoustically its implementation presents severe practical difficulties.
- o Intrinsic, or Skin-Longeron, Tuning. This concept did not work acoustically in the present study. If proven to function acoustically, its implementation raises many questions as to weight and how to ensure adequately tuned structures in production.

It is recommended that the promising sidewall concepts be further developed in two stages:

(1) Aircraft-application but flat panels should be built, and tested in a laboratory with a systematic variation of design parameters accompanied by appropriate theoretical analysis. By focusing on just the sidewalls found in this study to be promising, this stage of further development could lead to an optimization of acoustical parameters and a more accurate determination of the acoustical benefits offered by each concept.

(2) The one or two most promising sidewall designs identified above should then be built into an aircraft structure complete with all necessary nonacoustical features such as windows, airconditioning ducts and other attachments, and the sidewall effectiveness determined in ground tests in which structure adjacent to the test panel is made acoustically opaque.

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## APPENDIX A

### TEST PANEL DESCRIPTIONS

This Appendix contains design and test configuration details for the eleven panel types evaluated for their acoustical performance. Panels A to G are development panels while Panels H to K are reference panels. The details are presented in drawing form and are supplemented as appropriate by photographs.

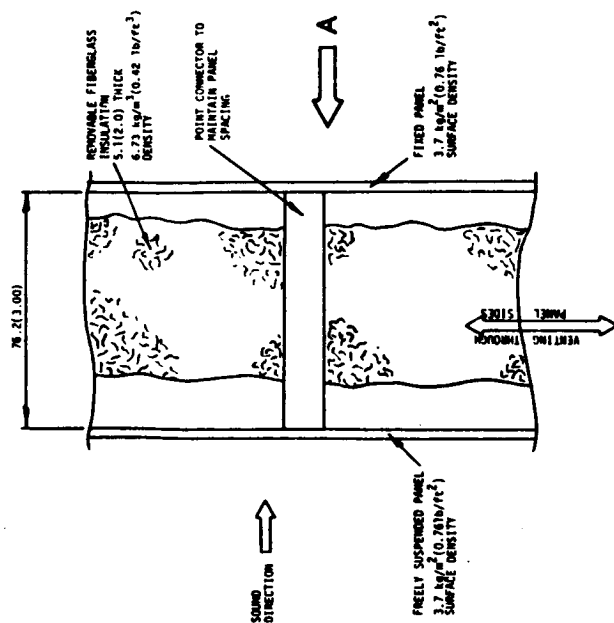
The details presented in these figures include:

- Panel size
- Panel construction
- Type of materials used
- Material thicknesses
- Material densities\*
- Configuration variations used in the acoustic tests

It should be noted that most of these panels were constructed using materials, or an orientation of materials or of the whole panel, that facilitated construction or testing. For example, Panel E was tested with the sound source on the "wrong" side to permit easy access for the purpose of adding damping, and Panel G was constructed by building a double wall structure on to the simplest-to-work side of a single wall panel.

\*Details of lead and longeron masses for Panel E are contained in Section 3.5 of the main report.

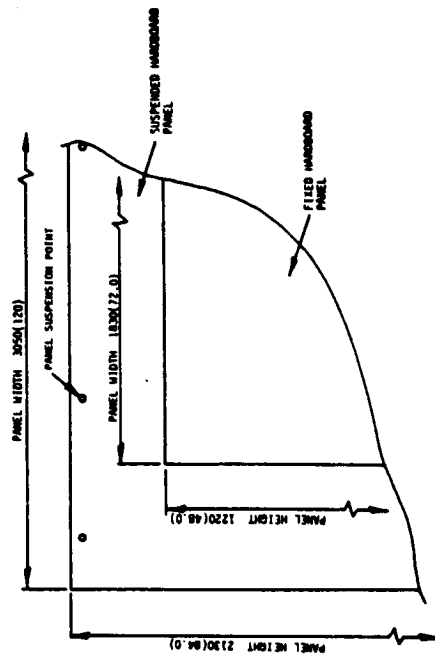
# CROSS SECTION



## CONFIGURATIONS

- A.1 - AS SHOWN ABOVE - WITHOUT FIBERGLASS INSULATION
- A.2 - AS A.1 BUT WITH FIBERGLASS INSULATION

# VIEW ON ARROW A

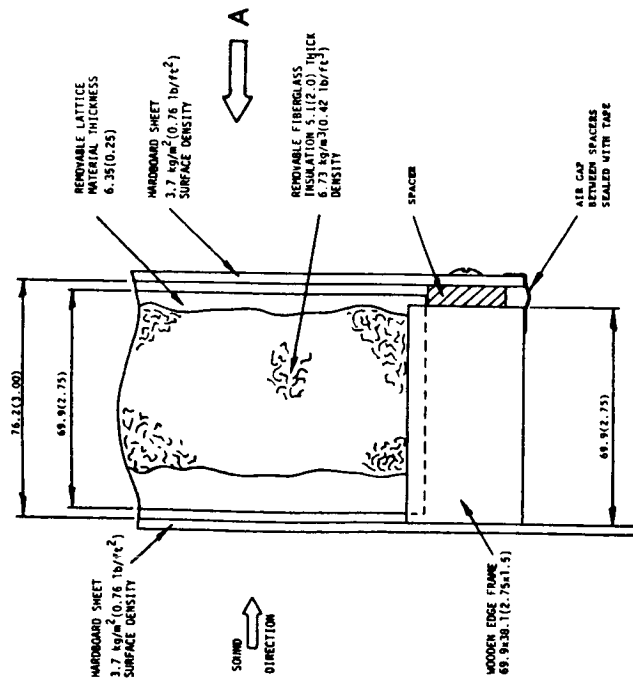


All dimensions in mm(in).

## PANEL A DOUBLE WALL WITH VENTED CAVITY

FIGURE A-1 DETAIL OF PANEL A

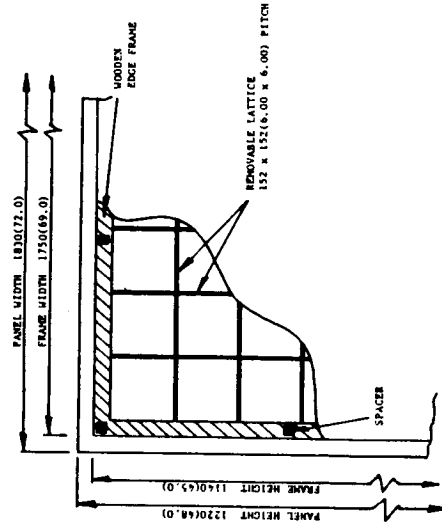
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## CONFIGURATIONS

- B.1 - AS SHOWN ABOVE - WITHOUT LATTICE AND FIBERGLASS INSULATION
- B.2 - AS B.1 BUT WITH FIBERGLASS INSULATION
- B.3 - AS B.1 BUT WITH LATTICE

# VIEW ON ARROW A



All dimensions in mm(in).

## PANEL B DOUBLE WALL WITH CAVITY LATTICE

FIGURE A-2 DETAIL OF PANEL B

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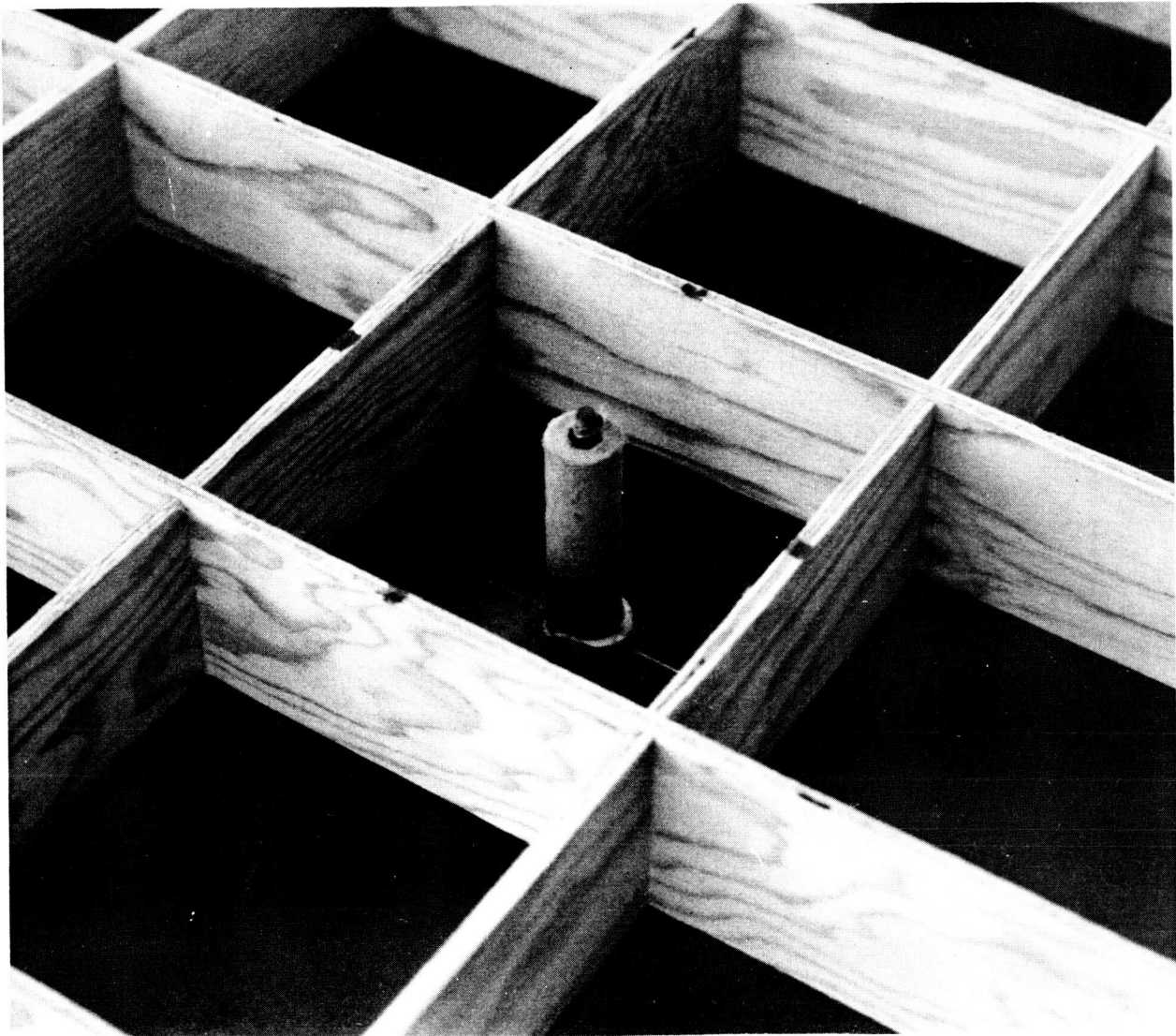
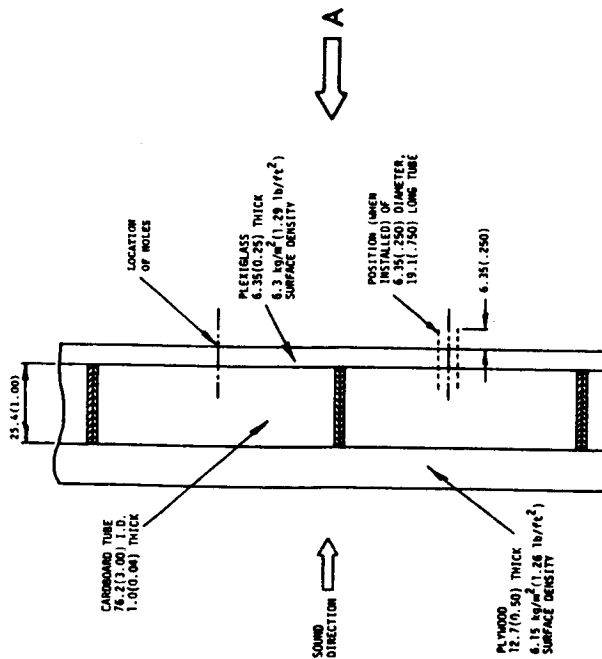


Figure A-3. Panel B Showing Lattice and Point Connector

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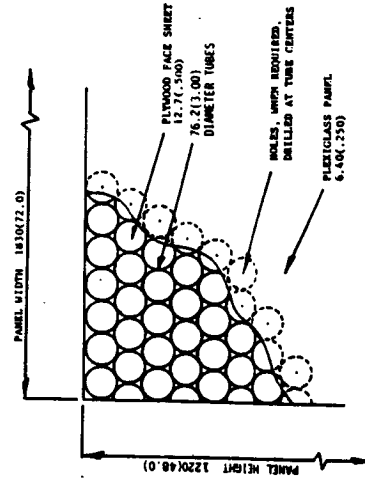
# CROSS SECTION



## CONFIGURATIONS

- C.1 - AS SHOWN ABOVE - NO HOLES
- C.2 - AS C.1 BUT WITH 3.18(.125) DIAMETER HOLE AT CENTER OF TUBES
- C.3 - AS C.2 BUT WITH 6.35(.250) DIAMETER HOLES
- C.4 - AS C.3 BUT WITH TUBES INSERTED IN HOLES
- C.5 - AS C.2 BUT WITH 12.7(.500) DIAMETER HOLES

# VIEW ON ARROW A



All dimensions in mm(1n).

## PANEL C WALL RESONATOR

FIGURE A-4 DETAIL OF PANEL C



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Figure A-5. Panel C Showing Test Configuration C.4

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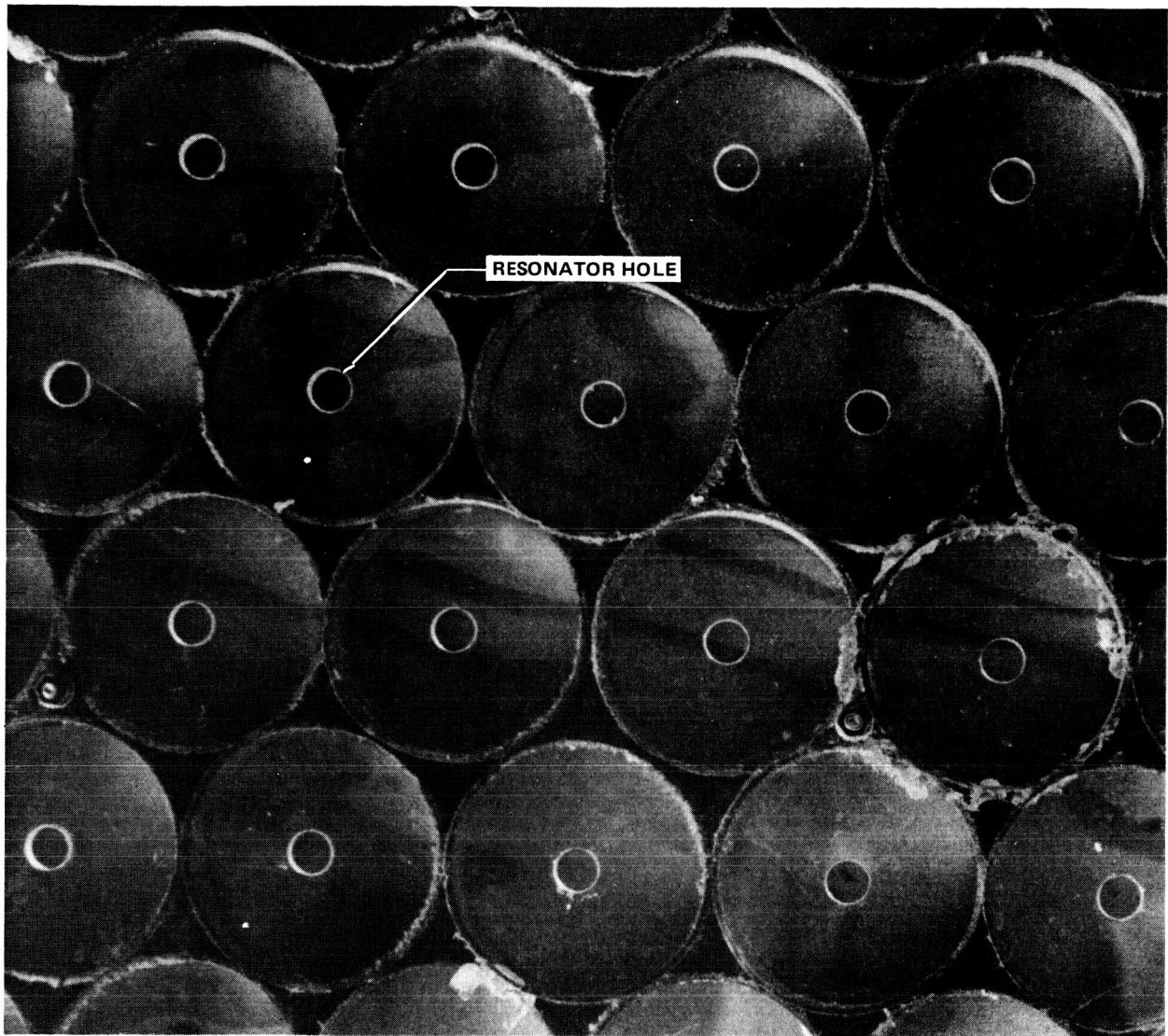
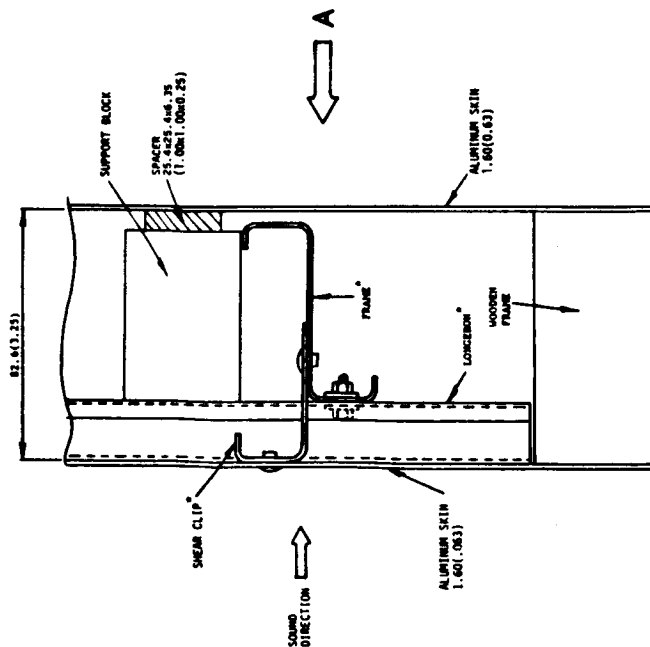


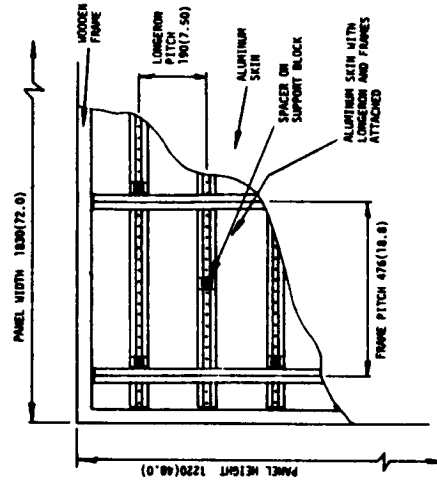
Figure A-6. Panel C Showing Test Configuration C.5

# CROSS SECTION



\*FOR DIMENSIONS SEE FIGURE A-21

# VIEW ON ARROW A



- CONFIGURATIONS**
- D.1 - AS SHOWN WITH NO CAVITY DECOMPRESSION
  - D.2 - AS D.1 BUT WITH  $2.1 \times 10^4 \text{ Pa}$  ( $3.0 \text{ lb/in}^2$ ) CAVITY DECOMPRESSION
  - D.3 - AS D.1 BUT WITH  $2.8 \times 10^4 \text{ Pa}$  ( $4.0 \text{ lb/in}^2$ ) CAVITY DECOMPRESSION
  - D.4 - AS D.1 BUT WITH  $3.5 \times 10^4 \text{ Pa}$  ( $5.0 \text{ lb/in}^2$ ) CAVITY DECOMPRESSION
  - D.5 - AS D.1 BUT WITH  $4.1 \times 10^4 \text{ Pa}$  ( $6.0 \text{ lb/in}^2$ ) CAVITY DECOMPRESSION
  - D.6 - AS D.1 BUT WITH  $4.8 \times 10^4 \text{ Pa}$  ( $7.0 \text{ lb/in}^2$ ) CAVITY DECOMPRESSION
  - D.7 - AS D.1 BUT WITH  $5.5 \times 10^4 \text{ Pa}$  ( $8.0 \text{ lb/in}^2$ ) CAVITY DECOMPRESSION
  - D.8 - AS D.1 BUT WITH  $6.2 \times 10^4 \text{ Pa}$  ( $9.0 \text{ lb/in}^2$ ) CAVITY DECOMPRESSION

All dimensions in mm(in).

## PANEL D DOUBLE WALL -- REDUCED CAVITY PRESSURE

FIGURE A-7 DETAIL OF PANEL D

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OF POOR QUALITY

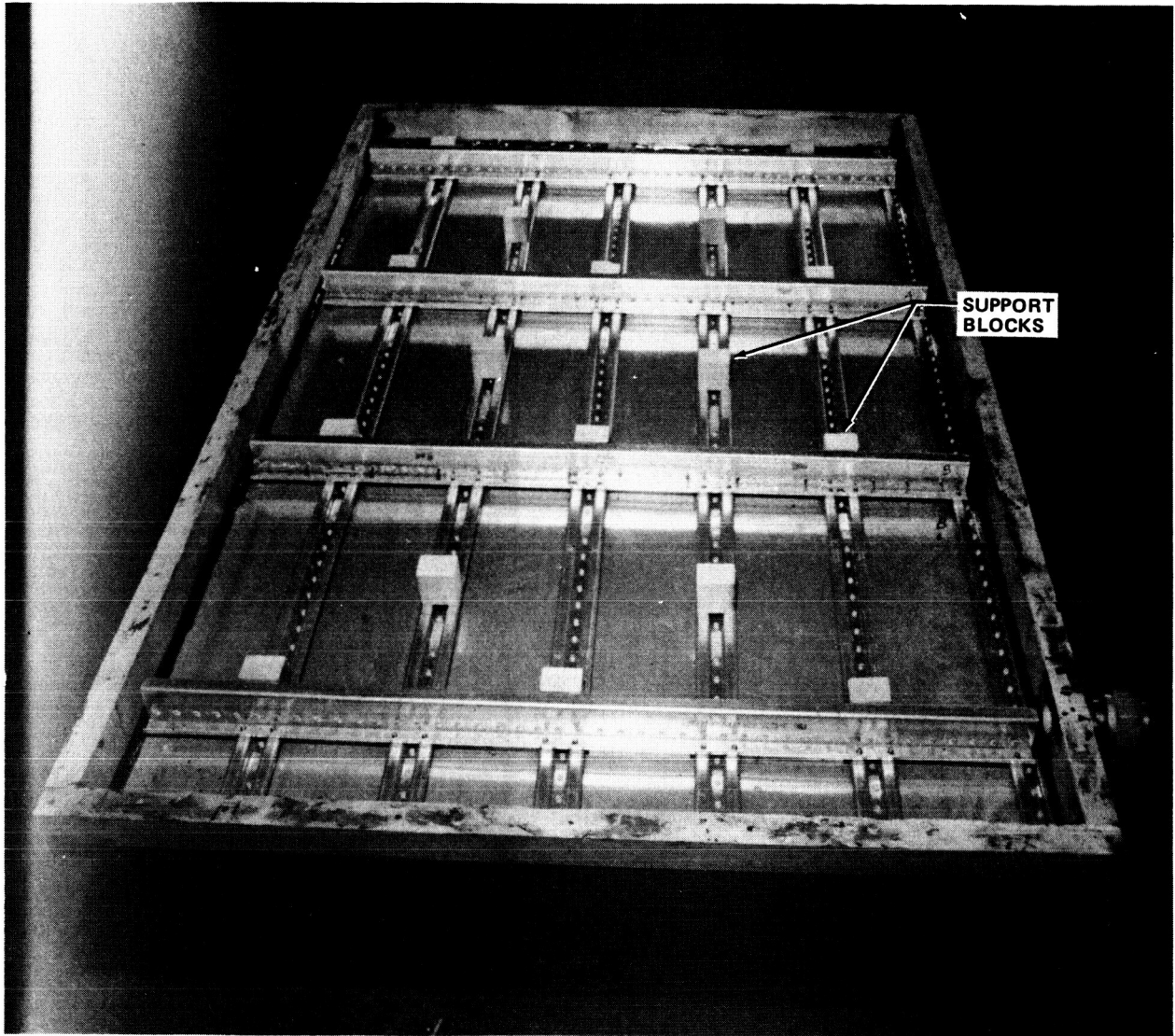
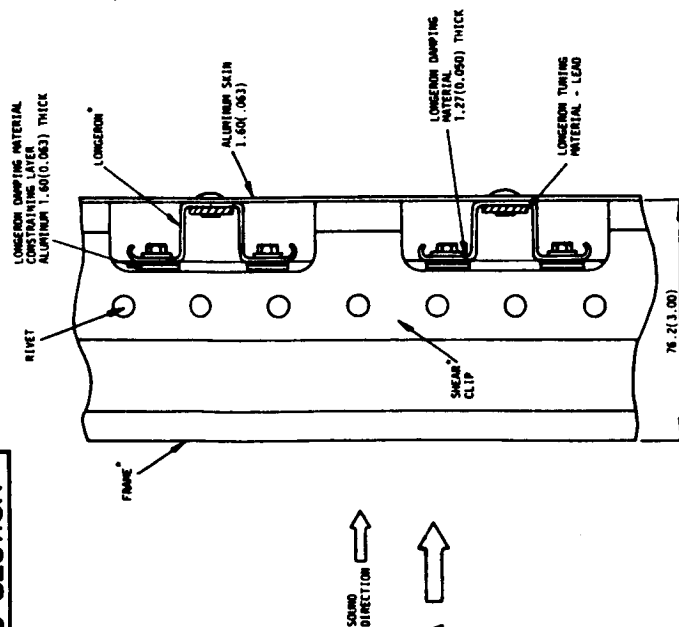


Figure A-8. Panel D Showing Location of Support Blocks before Application of Rubber Spacer and Second Skin

# CROSS SECTION

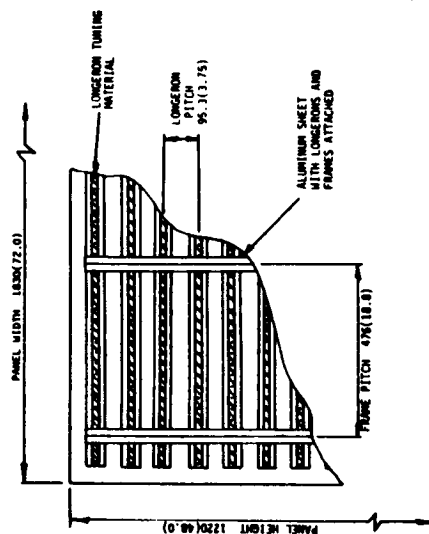


\*FOR DIMENSIONS SEE FIGURE A-21

## CONFIGURATIONS

- E.1 - AS CONFIGURATION K.1
- E.2 - AS E.1 BUT WITH MASS ON LONGERON FLANGES EQUIVALENT TO MASS OF DAMPING MATERIAL AND CONSTRAINING LAYER USED ON E.3, E.5 OR E.7.
- E.3 - AS E.2 BUT WITH DAMPING MATERIAL AND CONSTRAINING LAYER IN PLACE OF ADDED LONGERON FLANGE MASS.
- E.4 - AS E.2 BUT WITH MASS ADDED TO BASE OF LONGERONS TO ACHIEVE A "TUNED" STATE - SEE SECTION 3.5.
- E.5 - AS E.3 BUT WITH MASS ADDED TO BASE OF LONGERONS TO ACHIEVE A "TUNED" STATE - AS SHOWN ABOVE.
- E.6 - AS E.4 BUT WITH ABOUT TWICE THE MASS ADDED TO THE BASE OF THE LONGERONS - SEE SECTION 3.5.
- E.7 - AS E.5 BUT WITH ABOUT TWICE THE MASS ADDED TO THE BASE OF THE LONGERONS.

# VIEW ON ARROW A



All dimensions in mm(in).

## PANEL E SKIN-LONGERON TUNING

FIGURE A-9 DETAIL OF PANEL E



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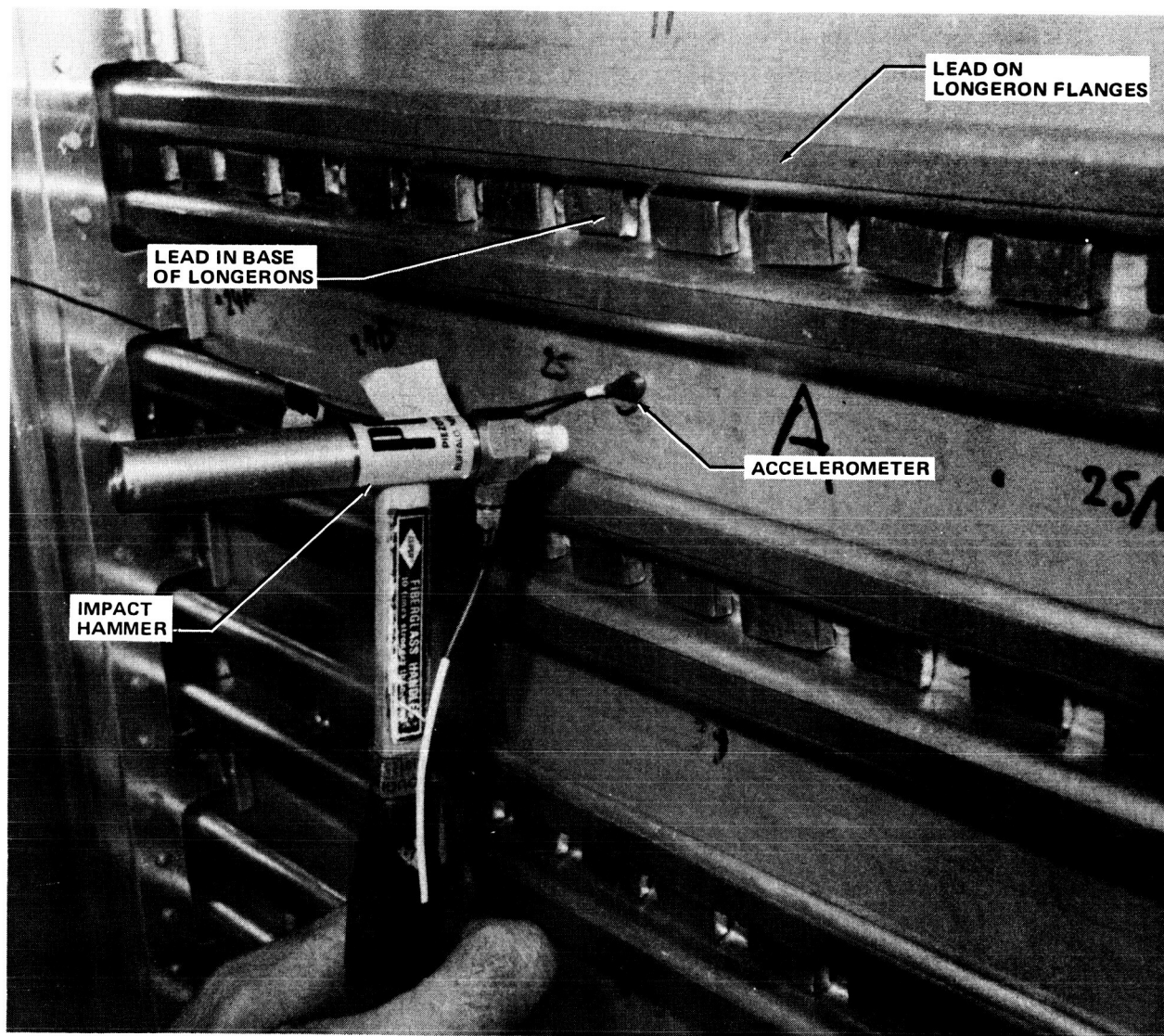


Figure A-10. Panel E Showing Method Used to Excite Panel during Panel Tuning

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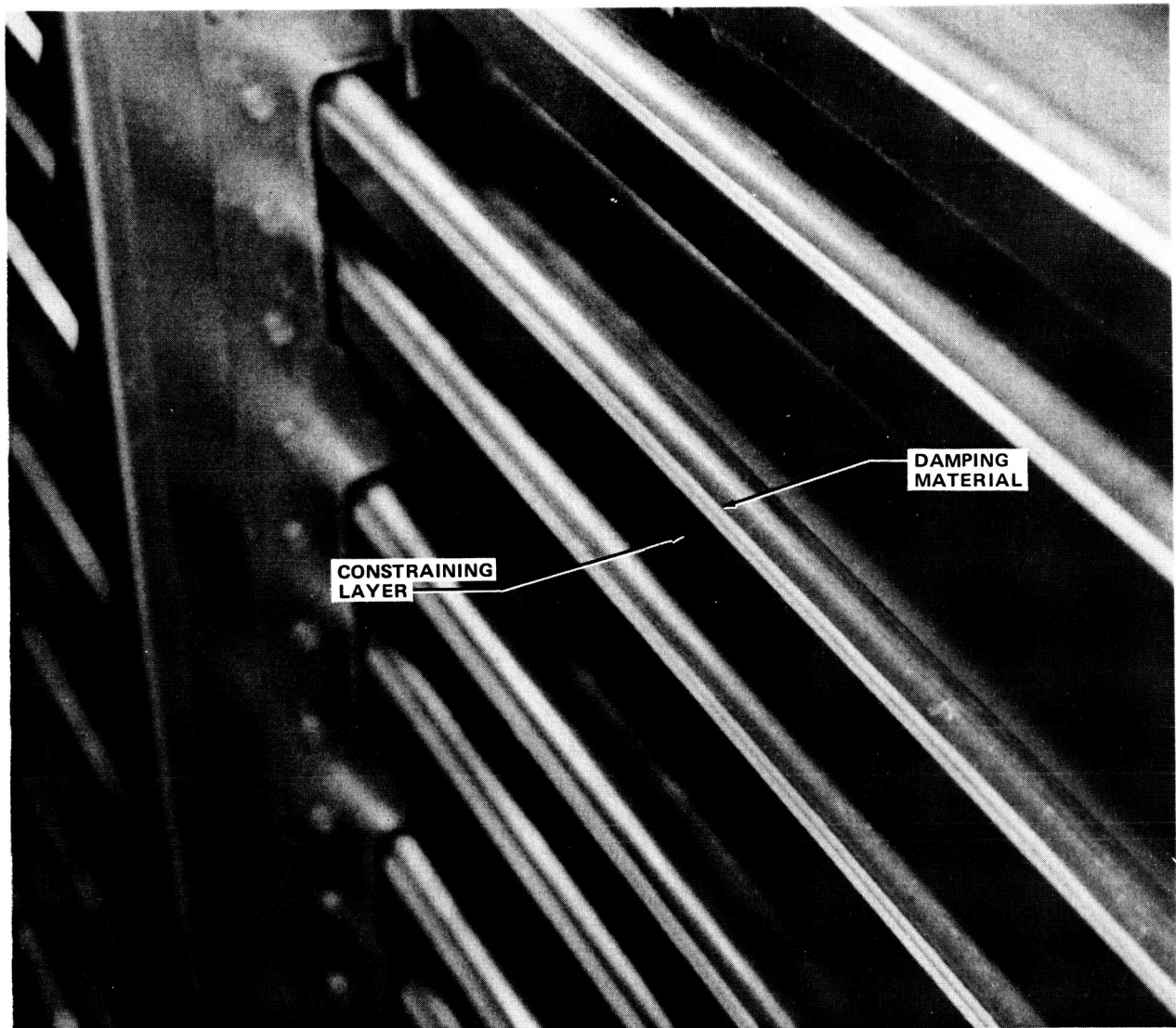
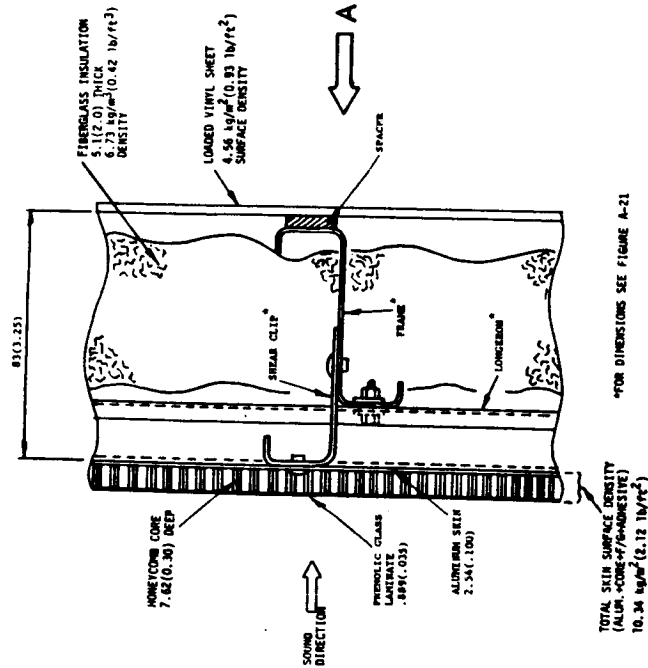


Figure A-11. Panel E Showing the Longeron Damping Material and Constraining Layer

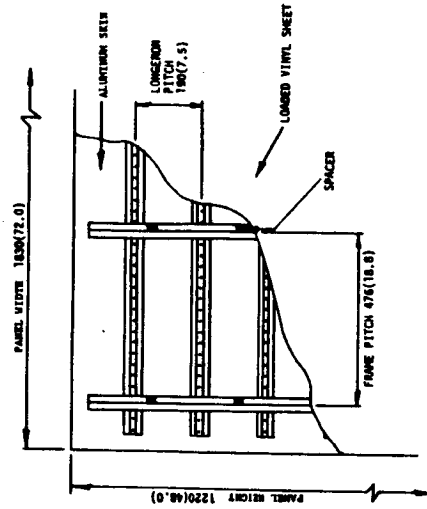
# CROSS SECTION



## CONFIGURATIONS

- F.1 - AS SHOWN ABOVE
- F.2 - AS SHOWN ABOVE BUT WITH LEAD VINYL SHEET CLOSE TO BUT NOT TOUCHING SPACER

# VIEW ON ARROW A



All dimensions in mm(in).

PANEL F DOUBLE WALL WITH STIFF PANEL

FIGURE A-12 DETAIL OF PANEL F

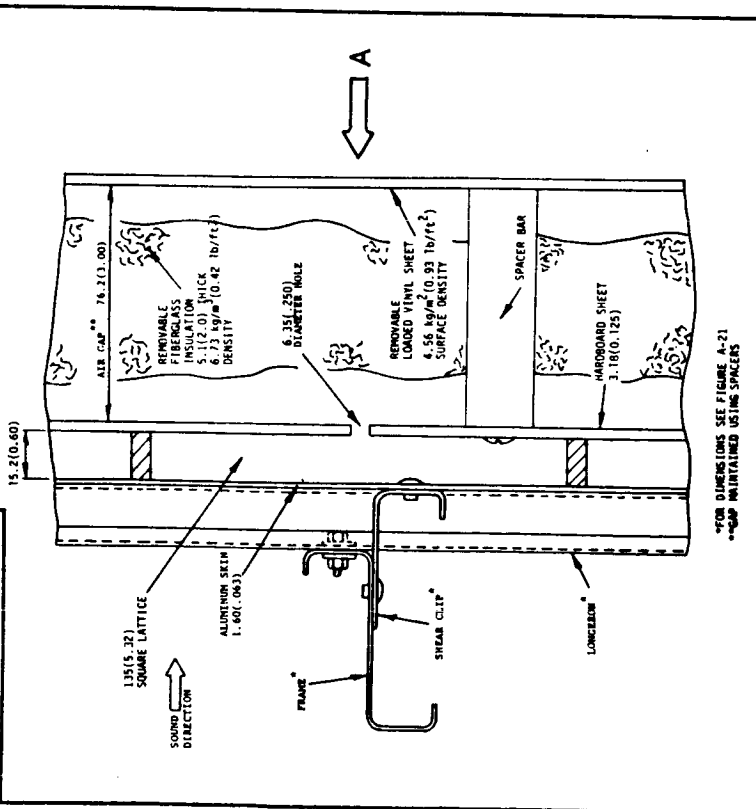


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Figure A-13. Panel F Showing Panel Assembly before Attaching the Loaded Vinyl Sheet

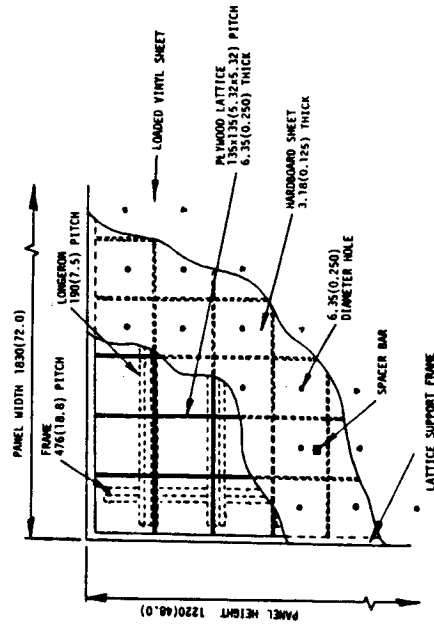
# CROSS SECTION



## CONFIGURATIONS

- G.1 - AS SHOWN ABOVE BUT WITHOUT LEAD VINYL SHEET AND FIBERGLASS INSULATION
- G.2 - AS SHOWN ABOVE WITH FIBERGLASS INSULATION

# VIEW ON ARROW A



All dimensions in mm(in).

## PANEL G DOUBLE WALL WITH RESONATORS

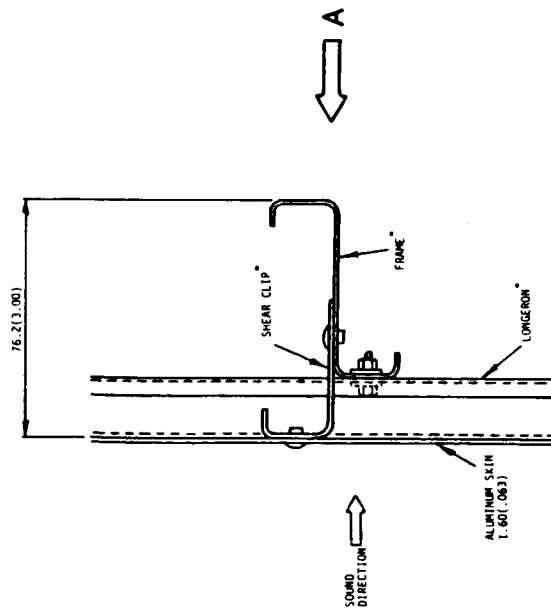
FIGURE A-14 DETAIL OF PANEL G

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Figure A-15. Panel G Showing Resonator Face Sheet and Spacer Bar

# CROSS SECTION

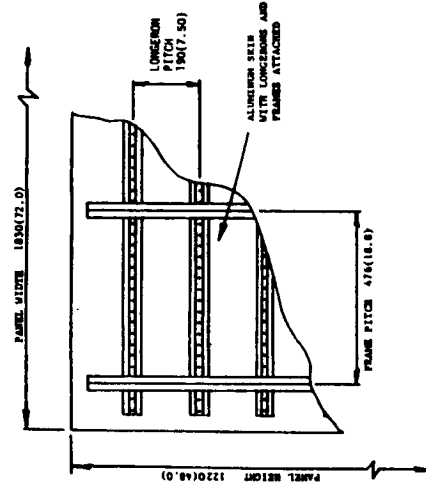


\*FOR DIMENSIONS SEE FIGURE A-21

## CONFIGURATIONS

- H.1 - AS SHOWN ABOVE
- H.2 - AS H.1 BUT WITH SOURCE DIRECTION FROM OPPOSITE SIDE

# VIEW ON ARROW A



All dimensions in mm(in).

PANEL H CONVENTIONAL SIDEWALL 1

FIGURE A-16 DETAIL OF PANEL H

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OF POOR QUALITY

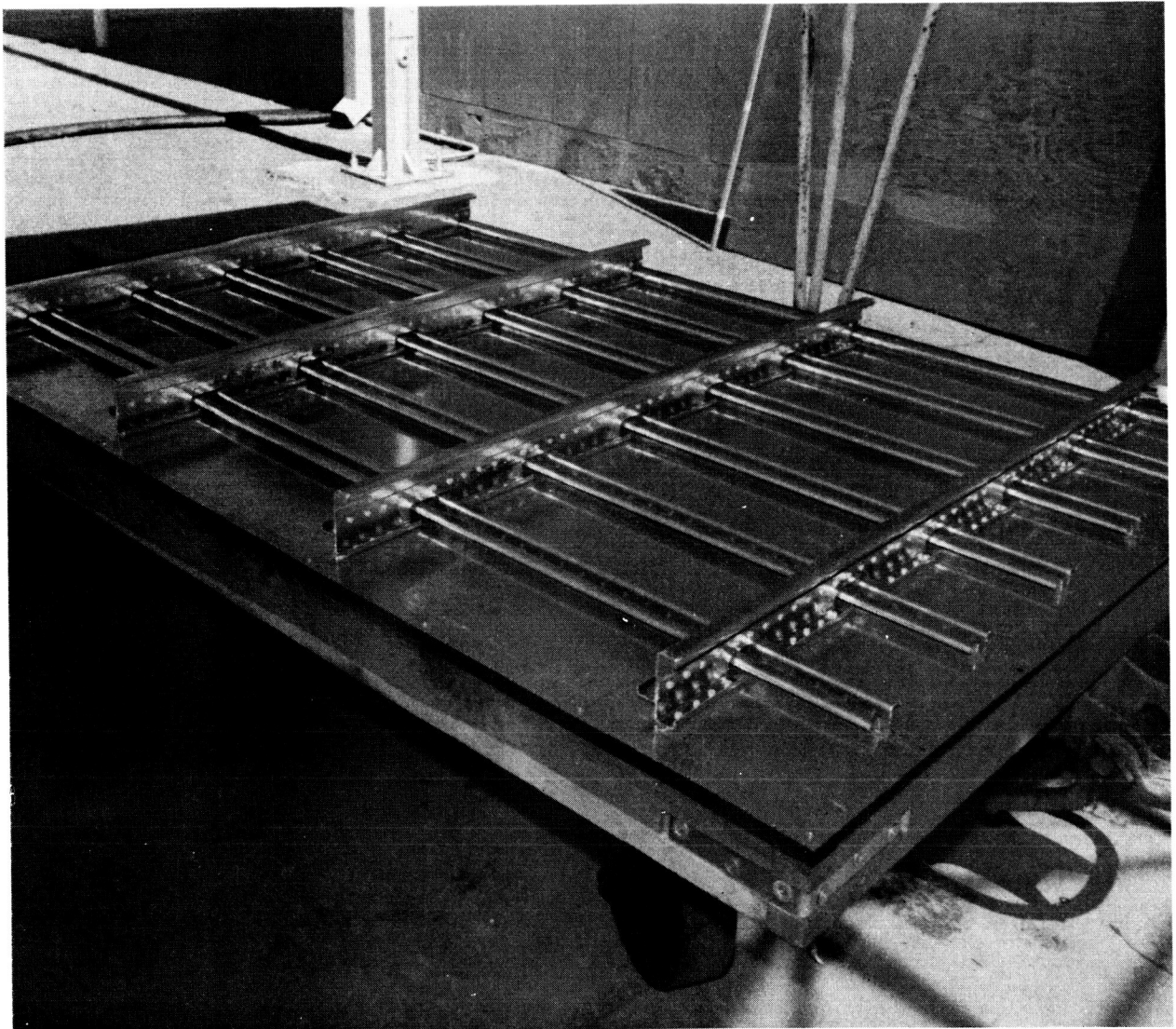
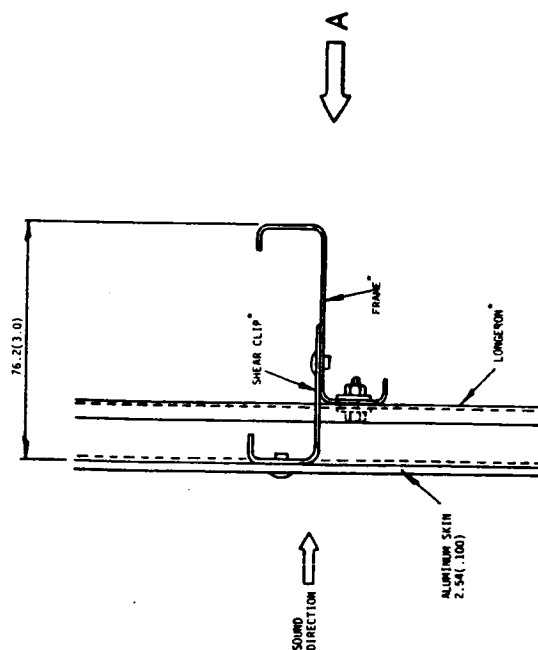


Figure A-17. Panel H Showing Conventional Skin, Longeron, Frame Construction

# CROSS SECTION

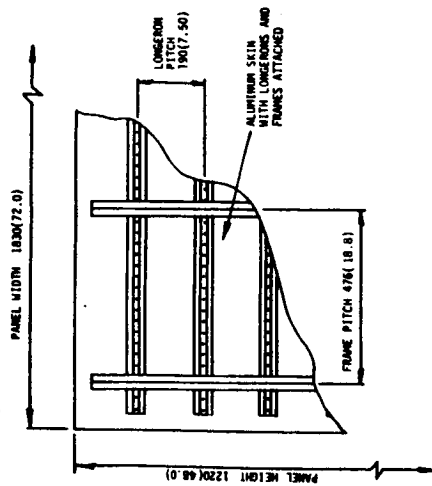


\*FOR DIMENSIONS SEE FIGURE A-21

## CONFIGURATIONS

I.1 - AS SHOWN ABOVE

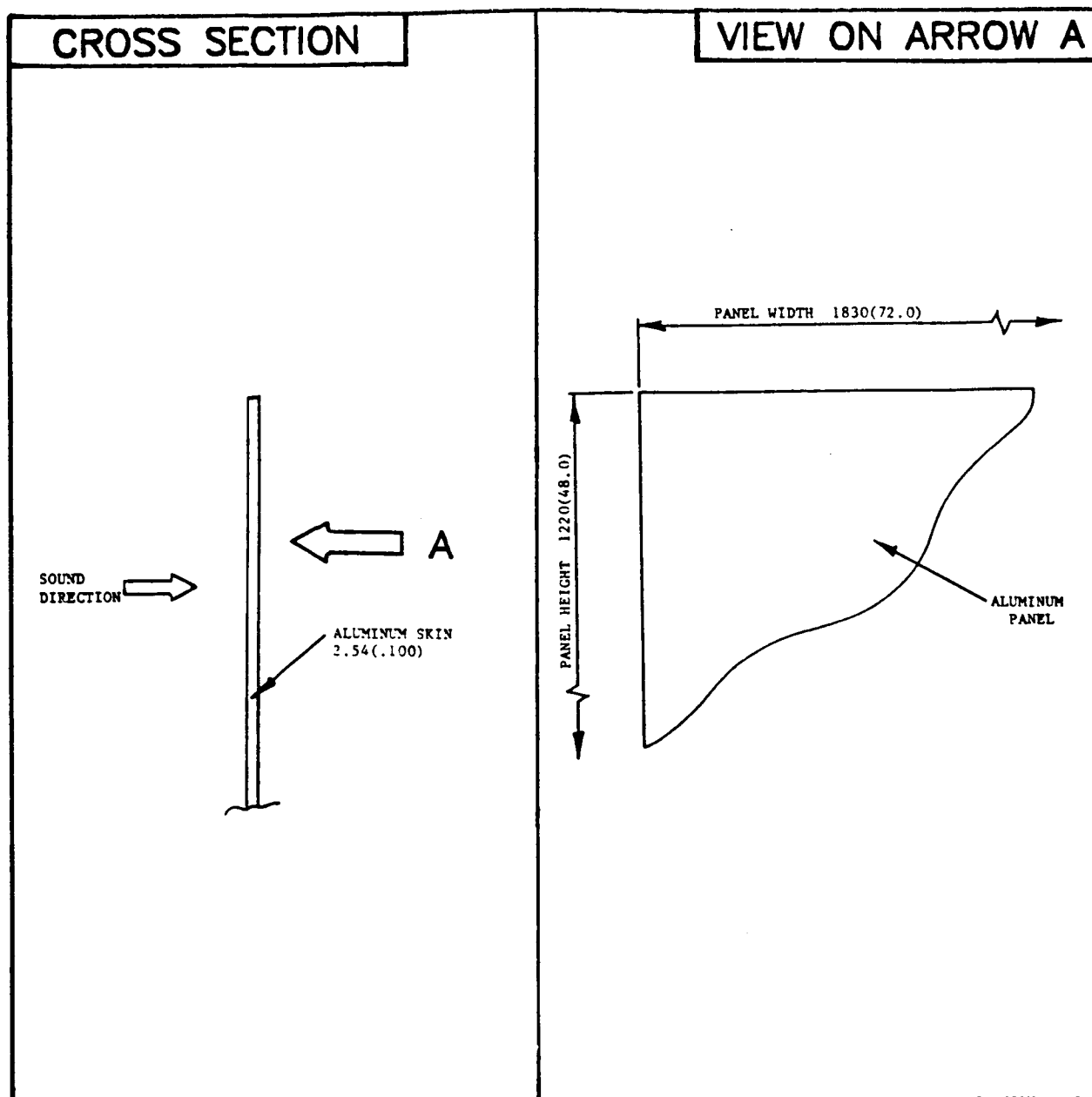
# VIEW ON ARROW A



All dimensions in mm(in).

PANEL I CONVENTIONAL SIDEWALL 2

FIGURE A-18 DETAIL OF PANEL I



## CONFIGURATIONS

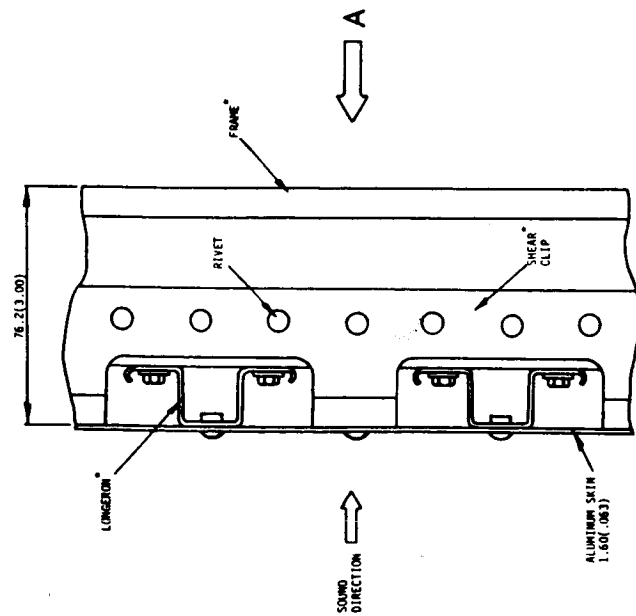
J.1 - AS SHOWN ABOVE

All dimensions in mm(in).

PANEL J SKIN ONLY

FIGURE A-19 DETAIL OF PANEL J

# CROSS SECTION

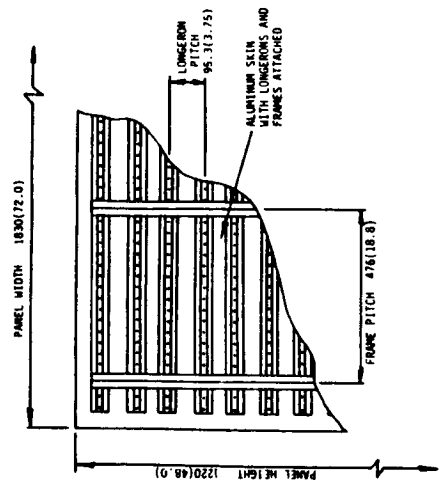


\*FOR DIMENSIONS SEE FIGURE A-21

## CONFIGURATIONS

K.1 - AS SHOWN ABOVE

# VIEW ON ARROW A



All dimensions in mm(in).

PANEL K CONVENTIONAL SIDEWALL - CLOSE LONGERON SPACING

FIGURE A-20 DETAIL OF PANEL K



## APPENDIX B

### ACOUSTIC TEST RESULTS

This Appendix contains plots of noise reduction vs frequency (on both log and linear scales) for each of the panel configurations tested. Omitted are plots for panel configurations D.4 through D.8. As noted in Section 5 of the main report, these results are virtually identical to those for D.2 and D.3.

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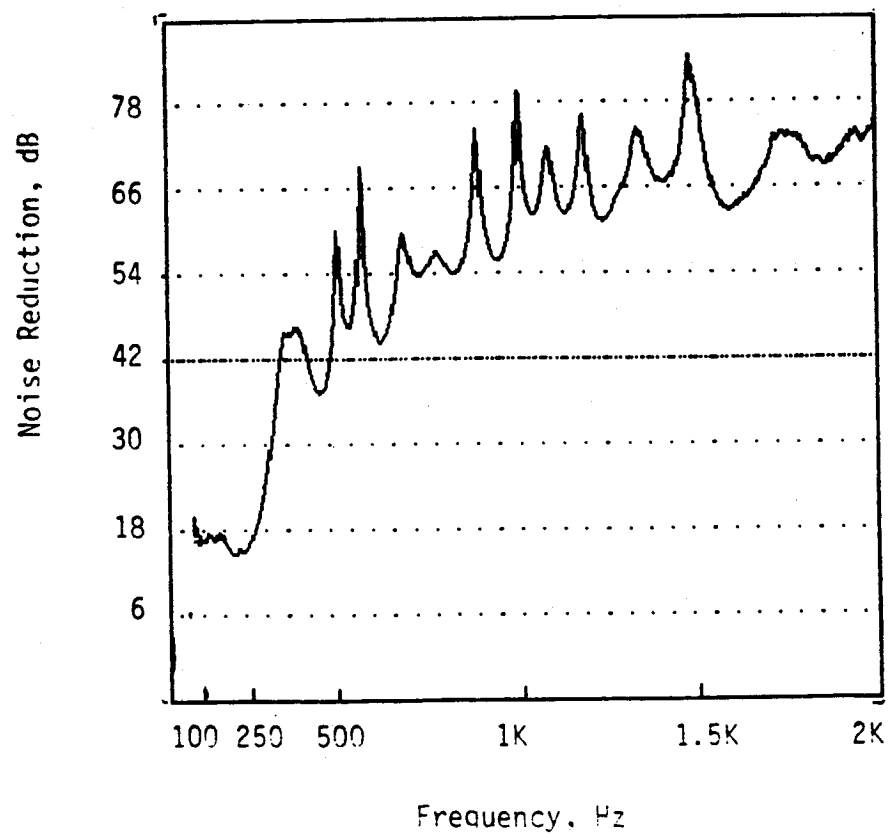
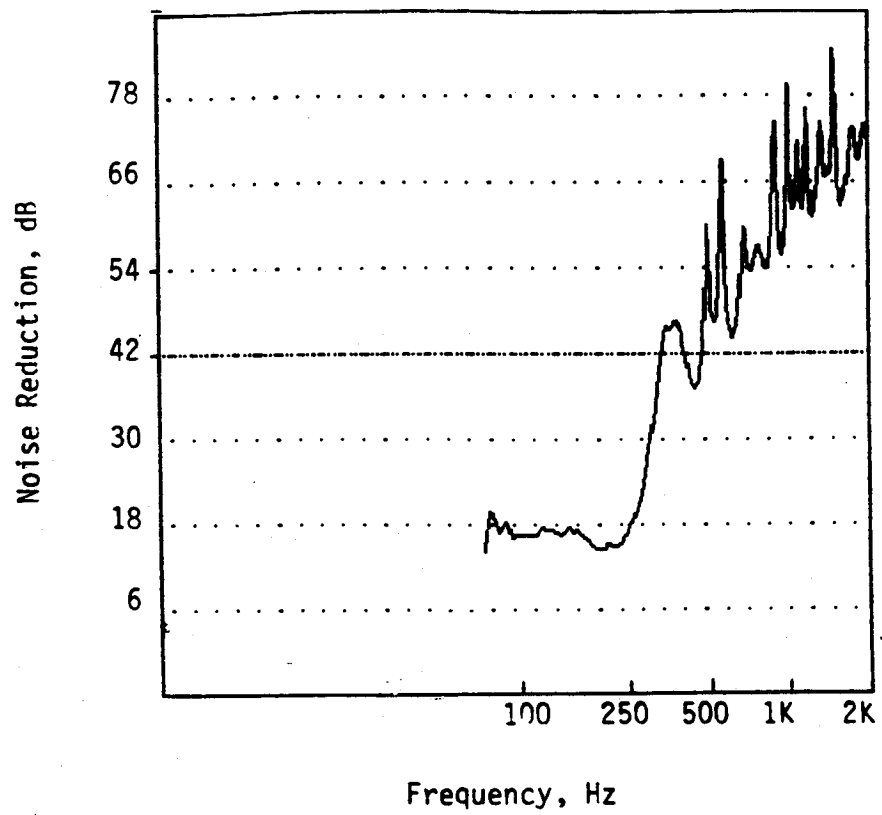


Figure B-1. Noise Reduction of Panel A.1

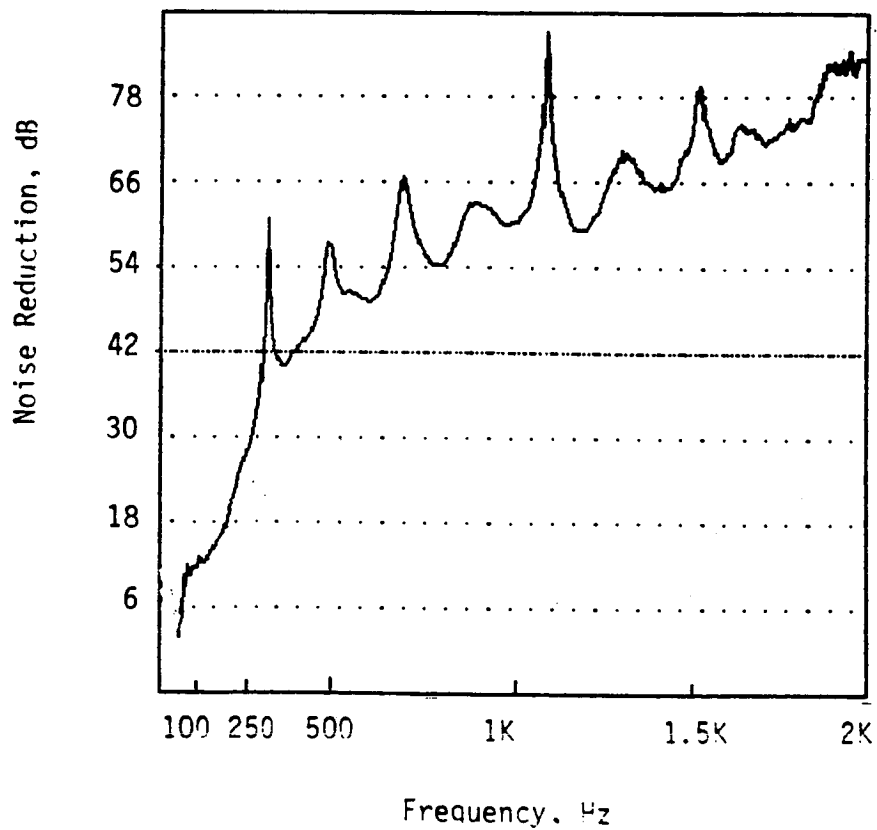
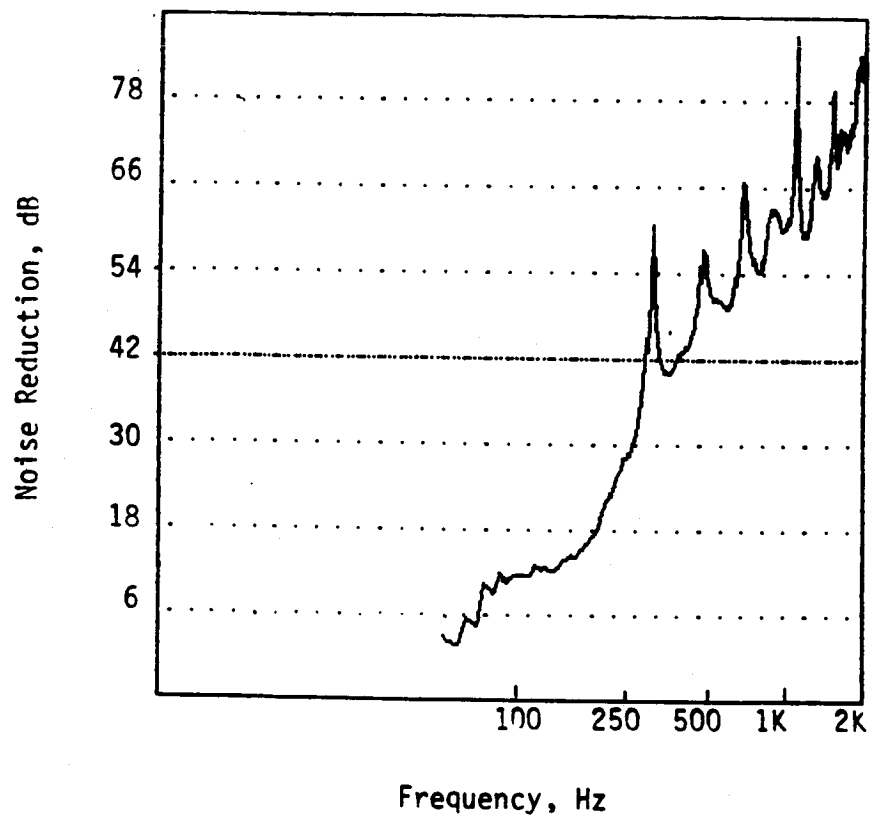


Figure B-2. Noise Reduction of Panel A.2

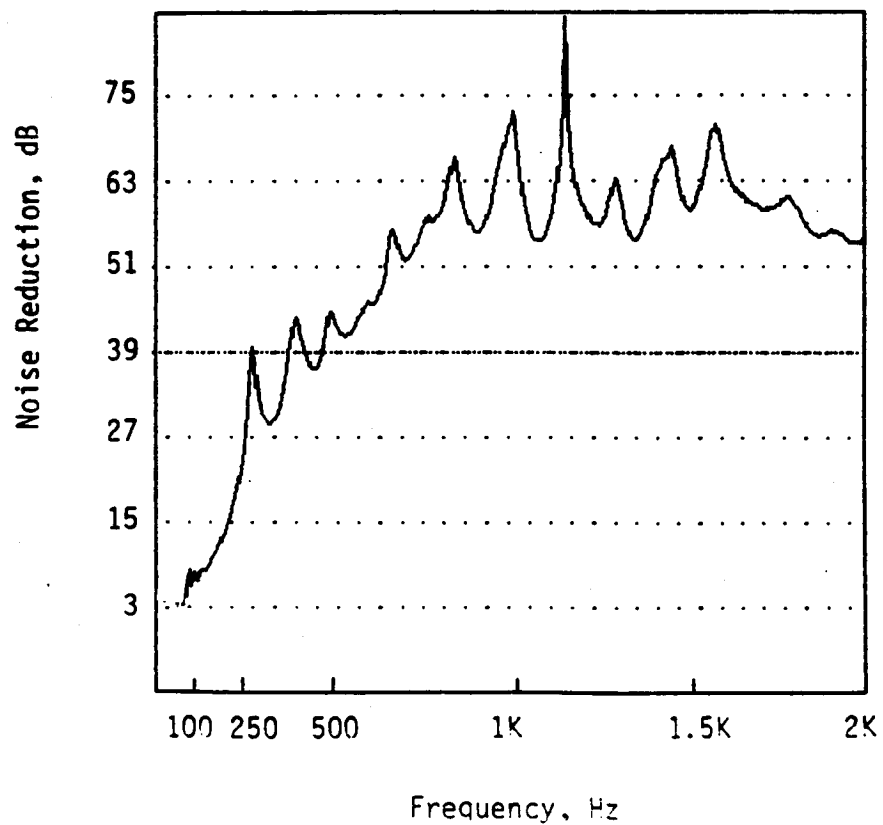
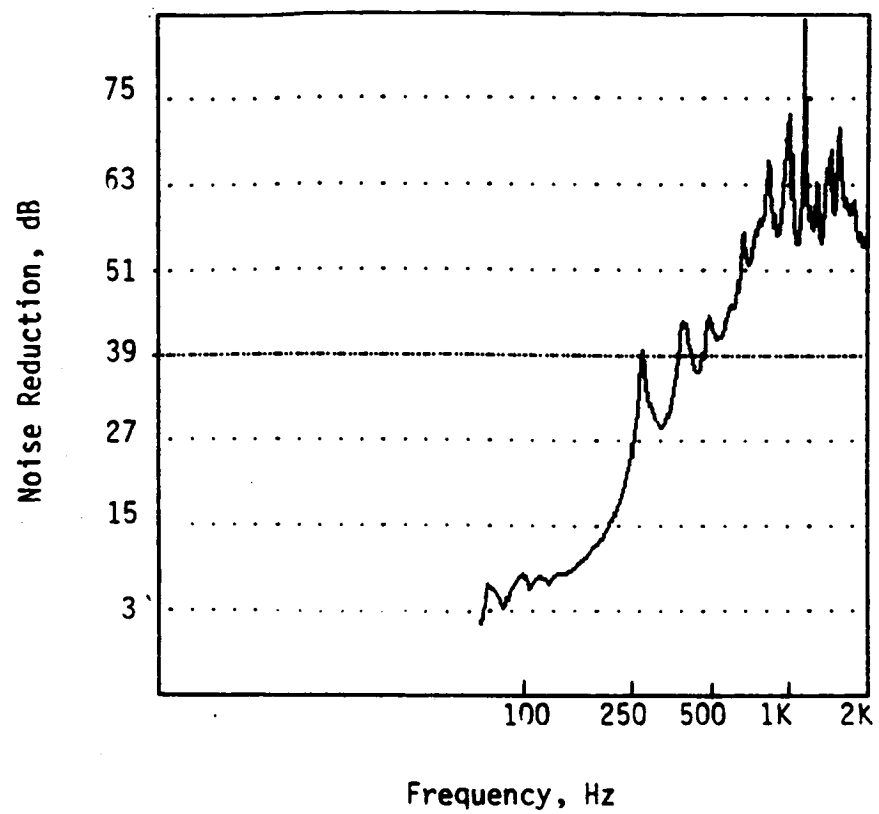


Figure B-3. Noise Reduction of Panel B.1

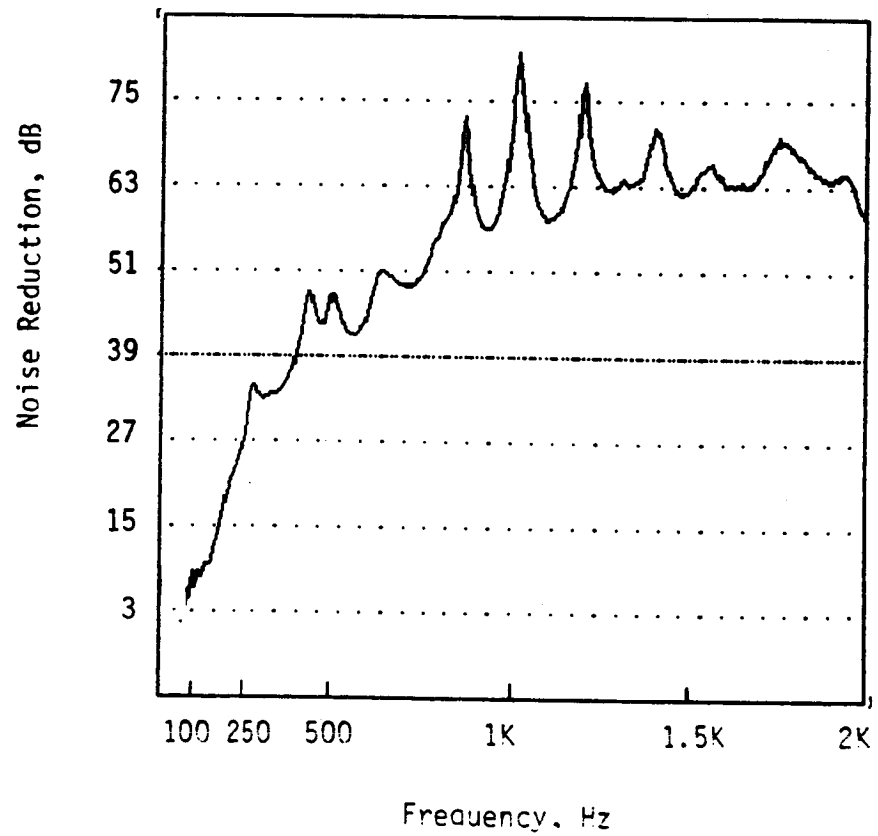
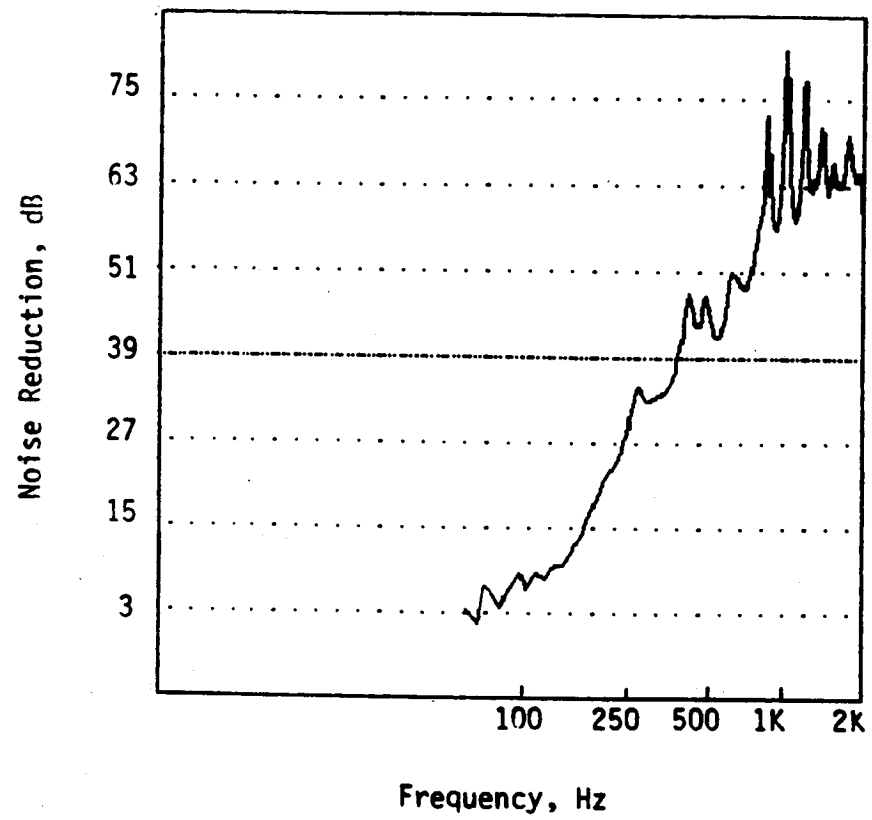


Figure B-4. Noise Reduction of Panel B.2

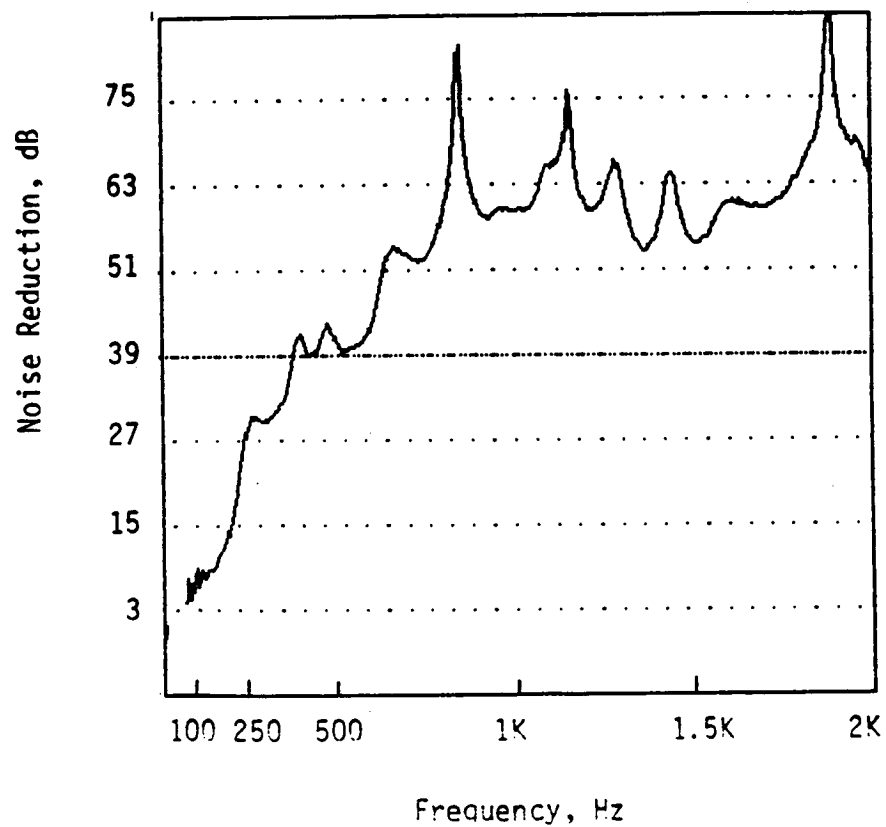
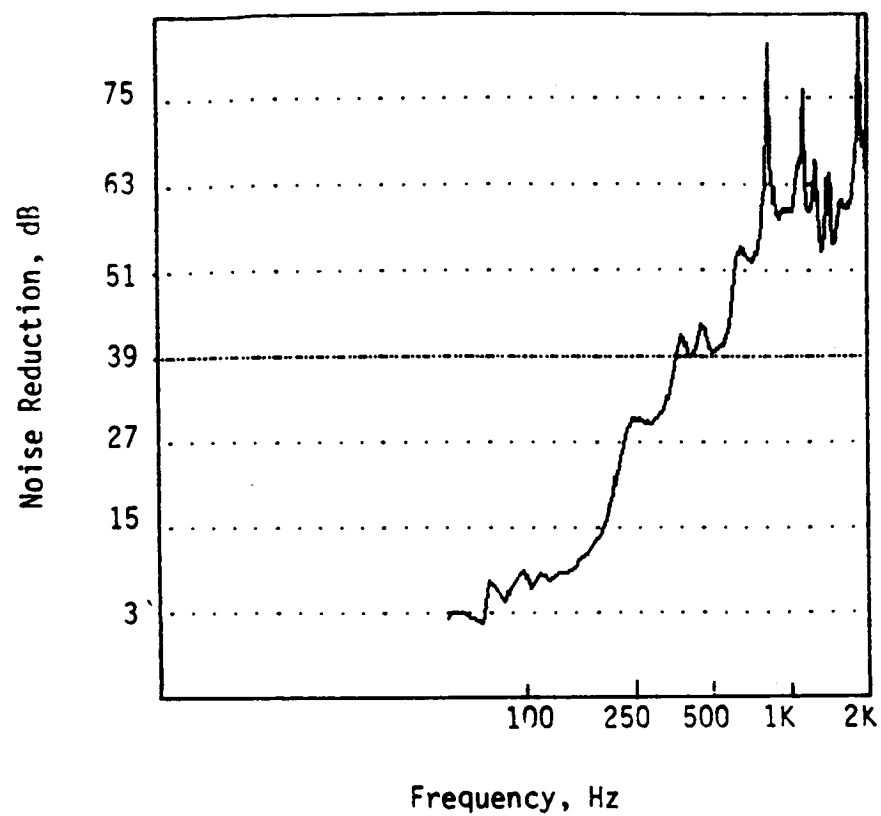


Figure B-5. Noise Reduction of Panel B.3

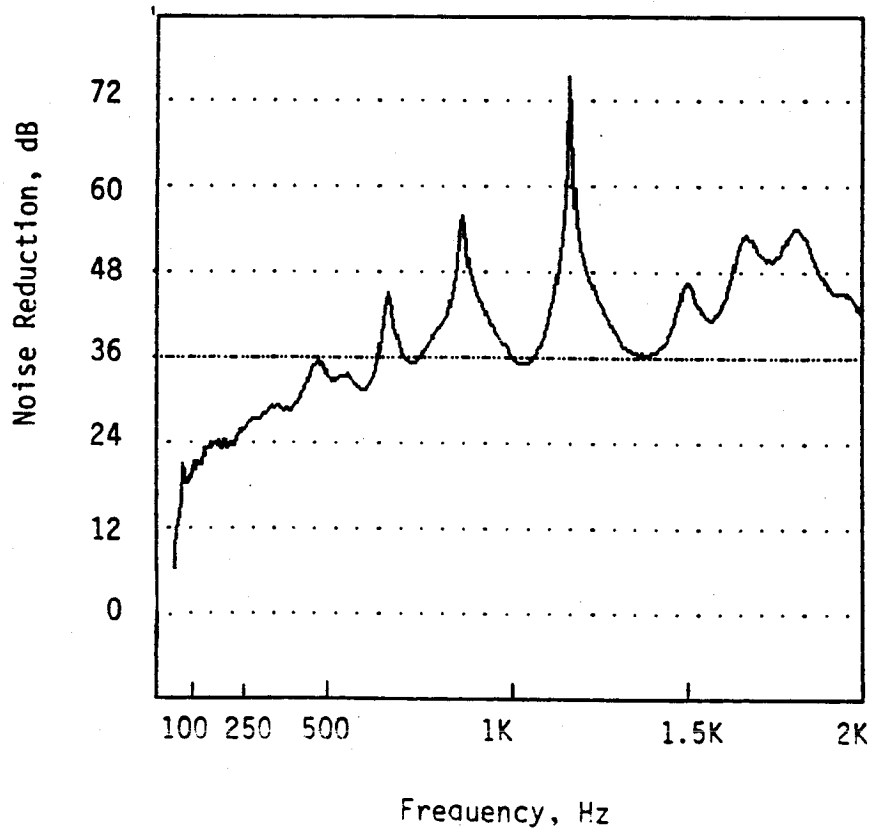
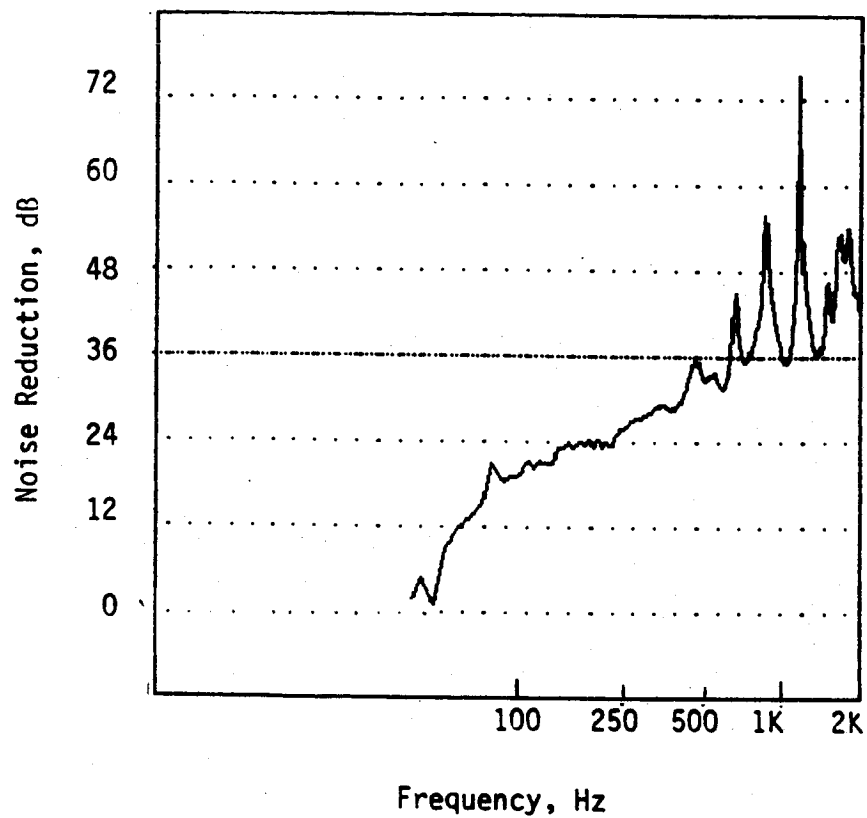


Figure B-6. Noise Reduction of Panel C.1

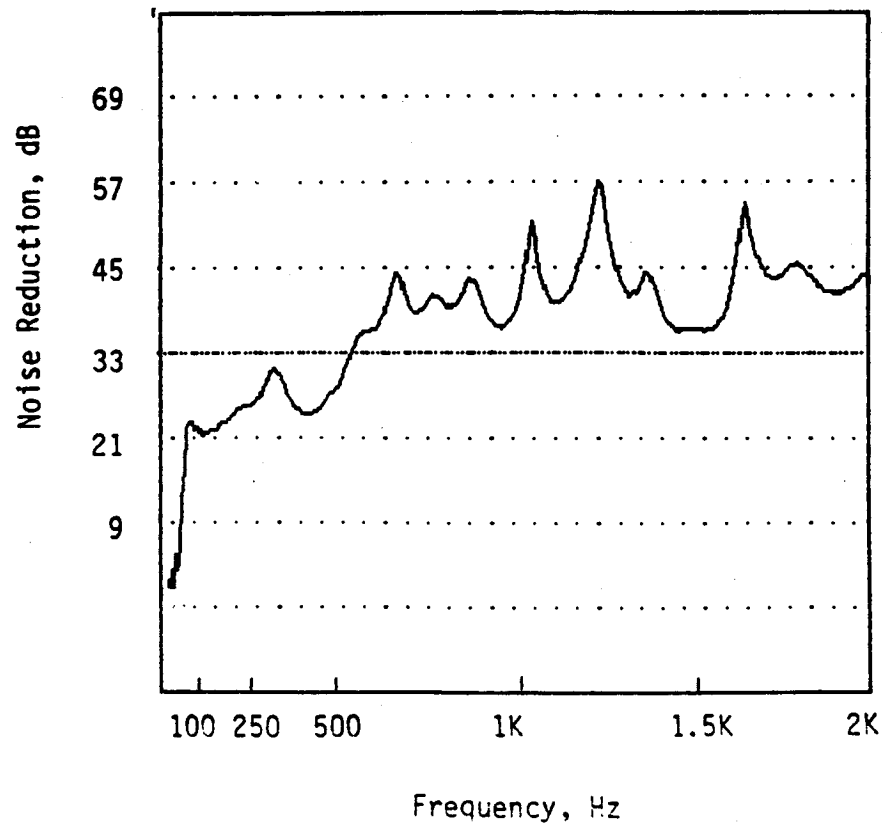
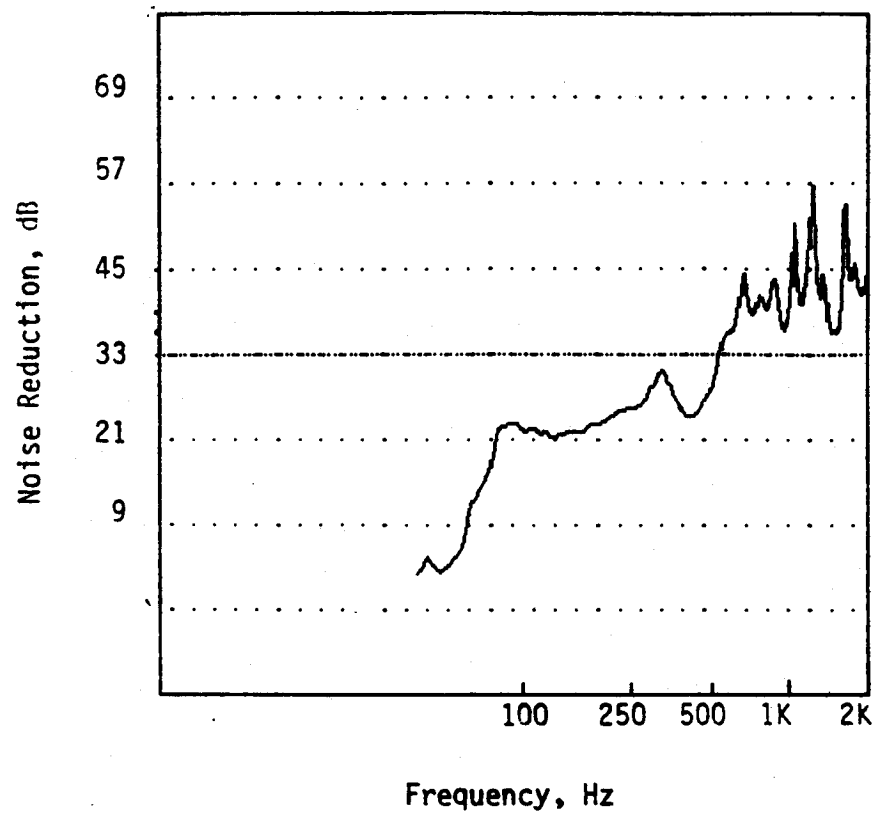


Figure B-7. Noise Reduction of Panel C.2



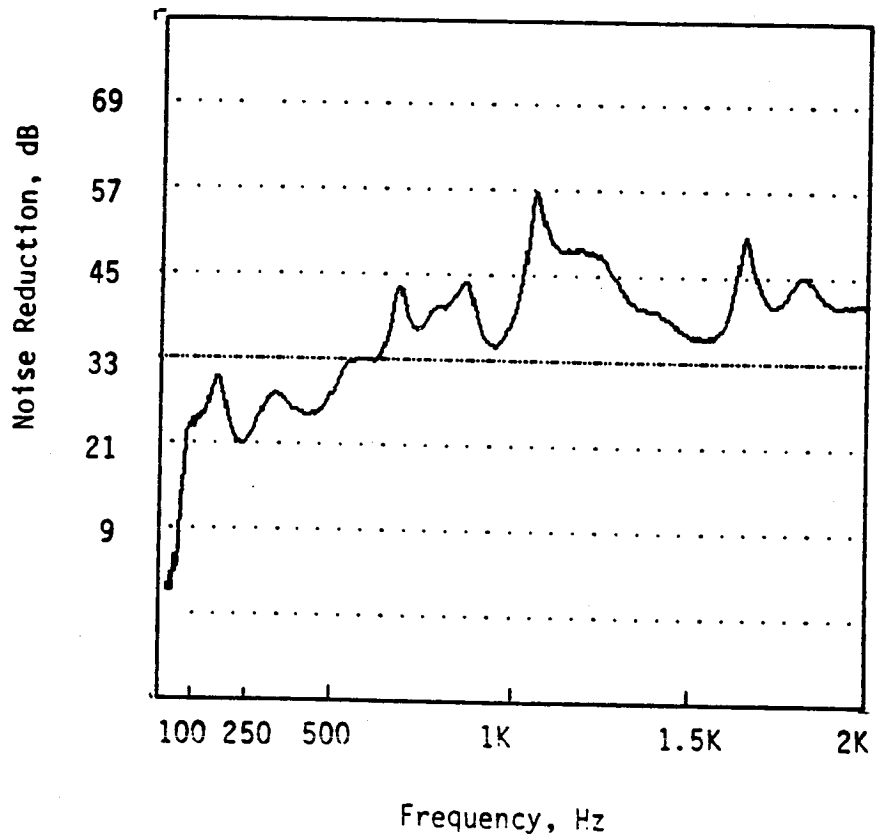
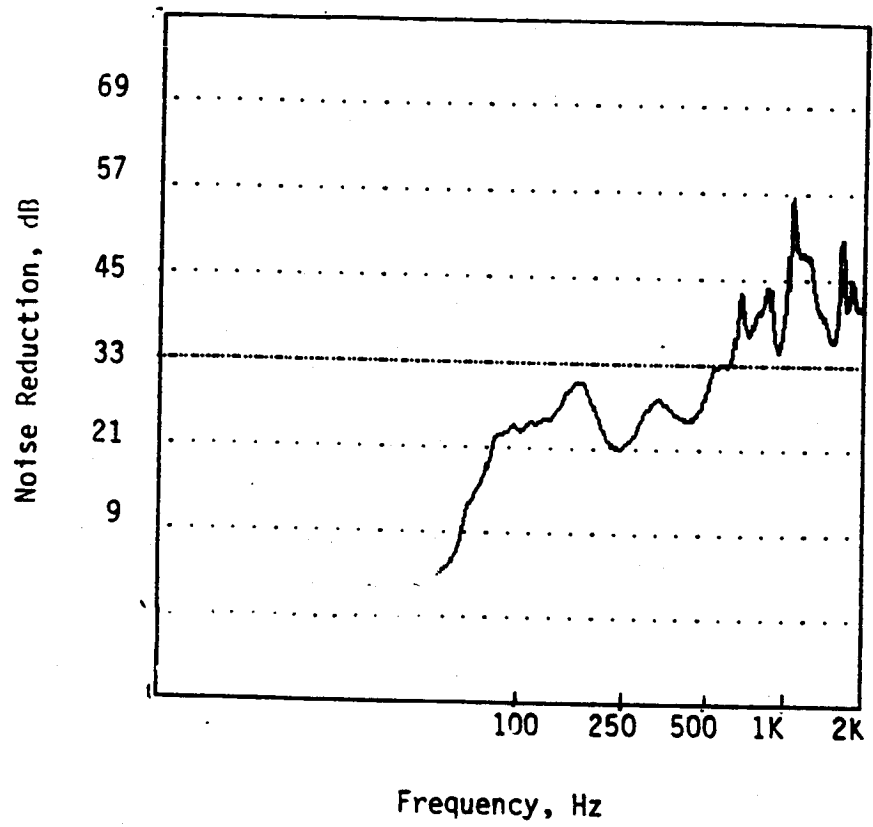


Figure B-8. Noise Reduction of Panel C.3

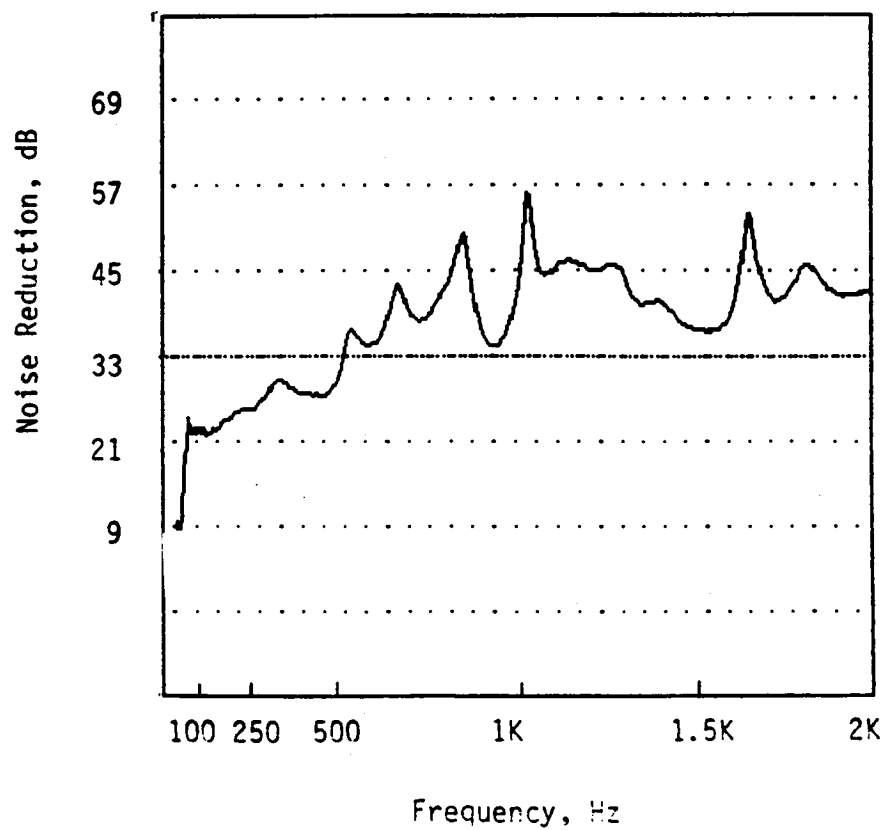
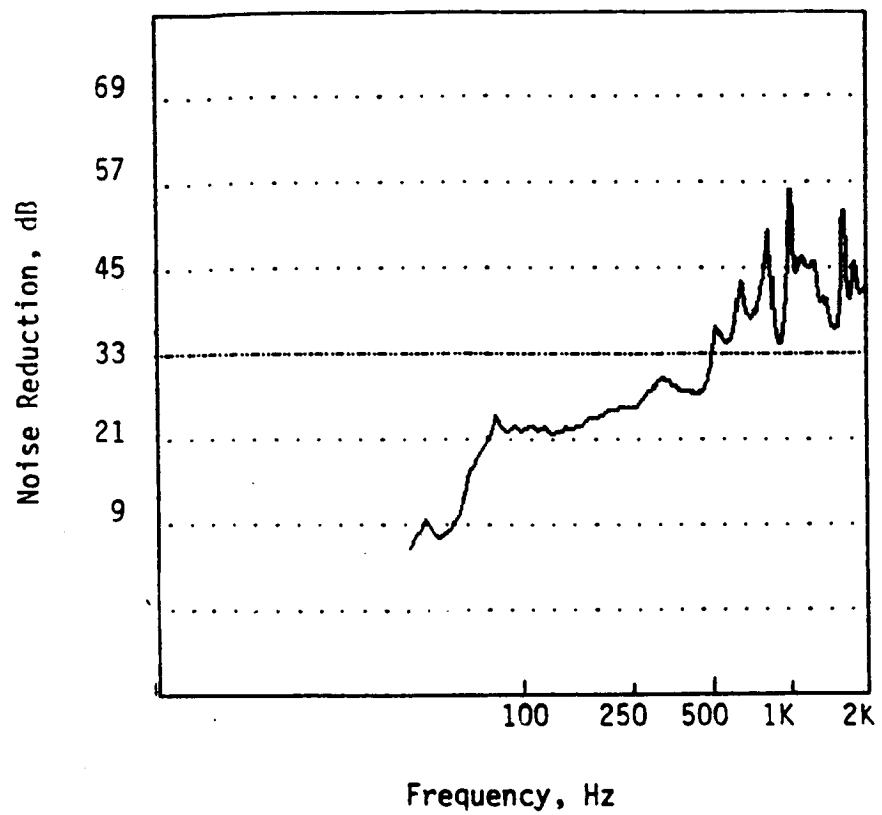


Figure B-9. Noise Reduction of Panel C.4

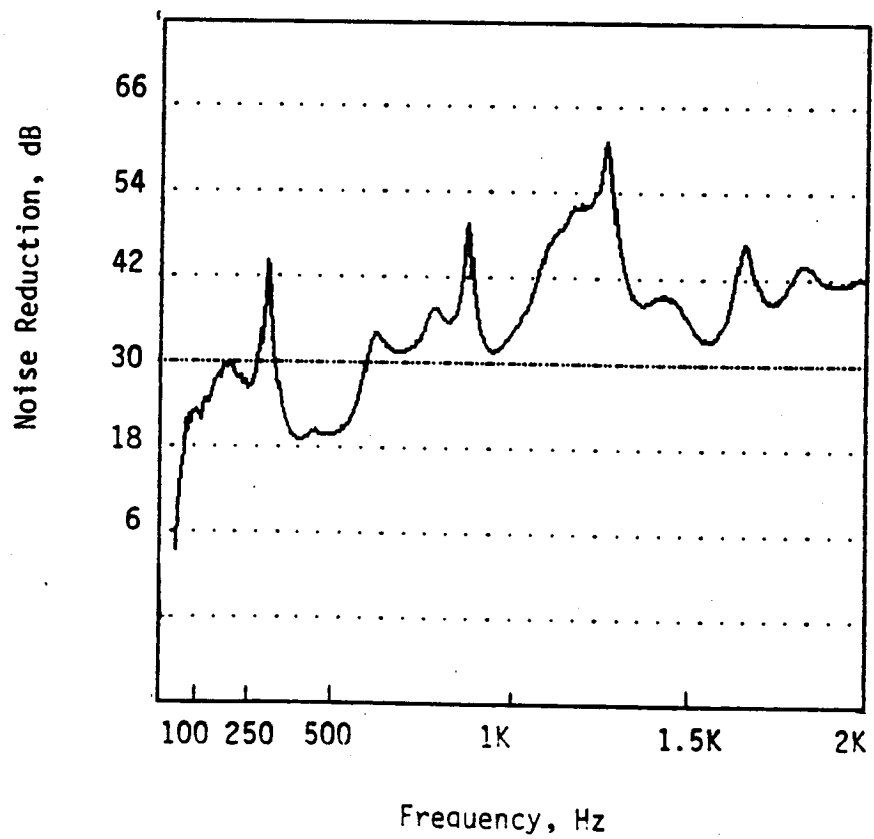
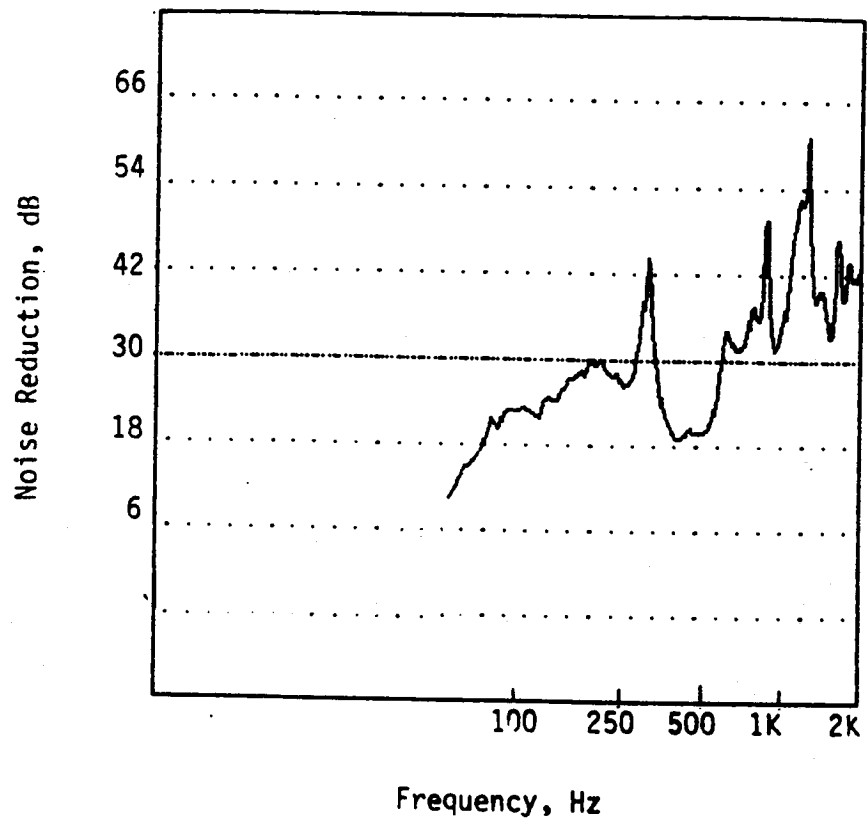


Figure B-10. Noise reduction of Panel C.5

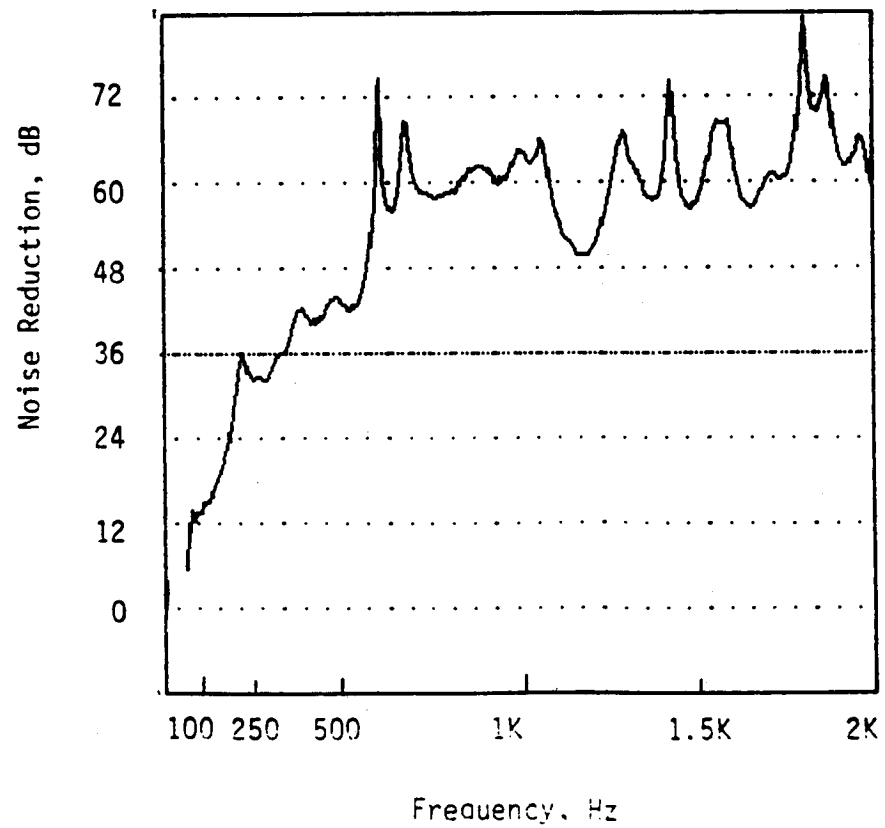
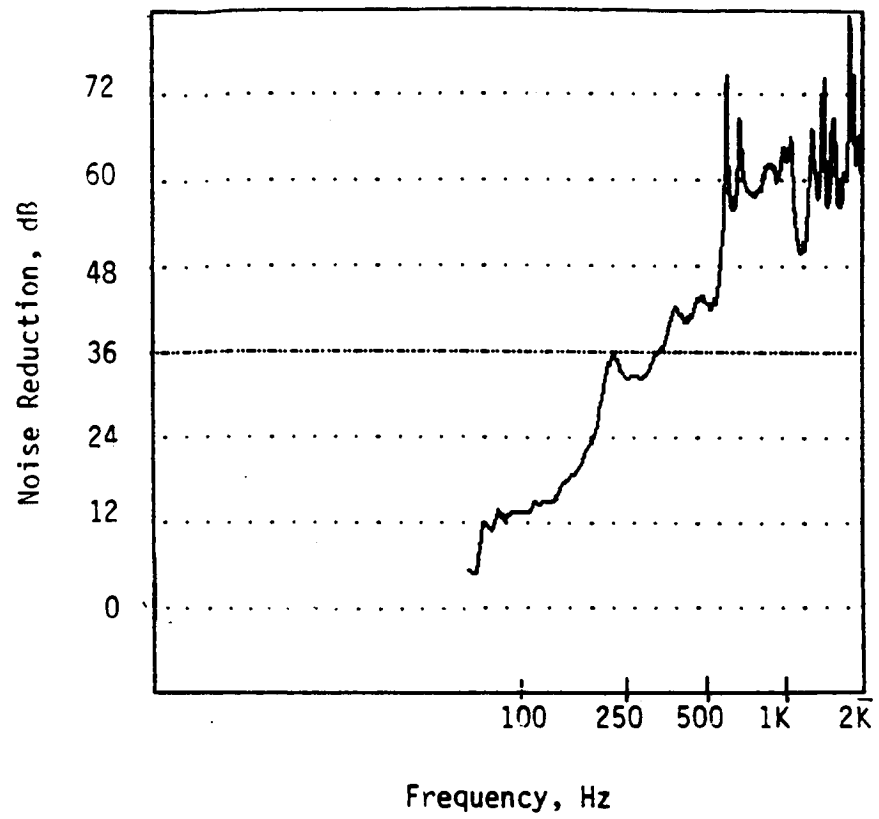


Figure B-11. Noise Reduction of Panel D.1

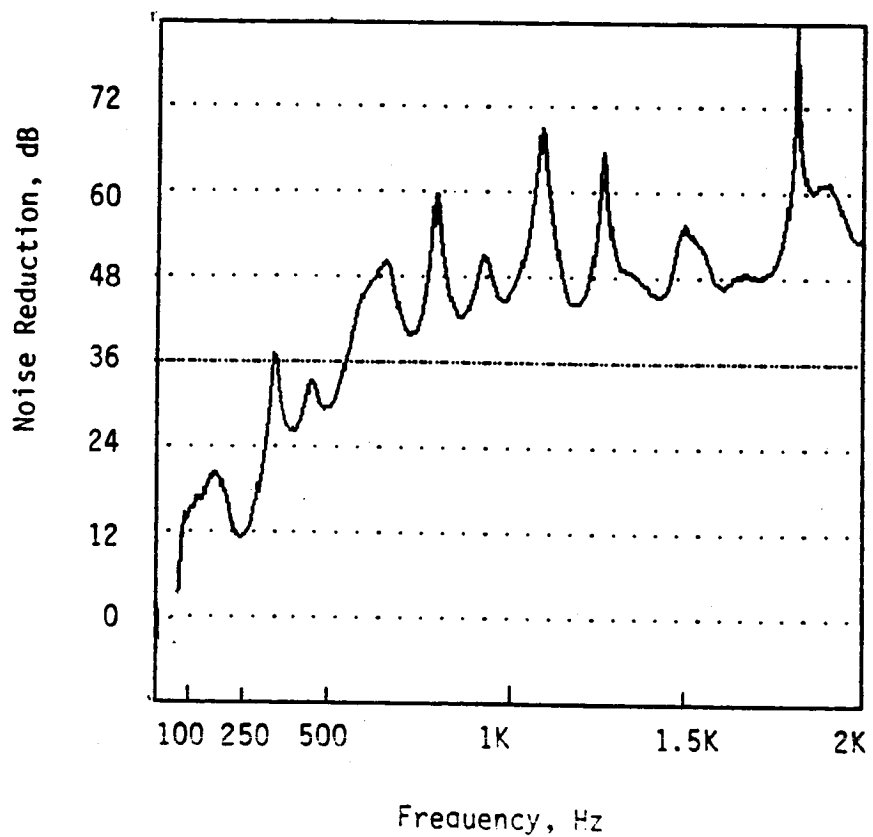
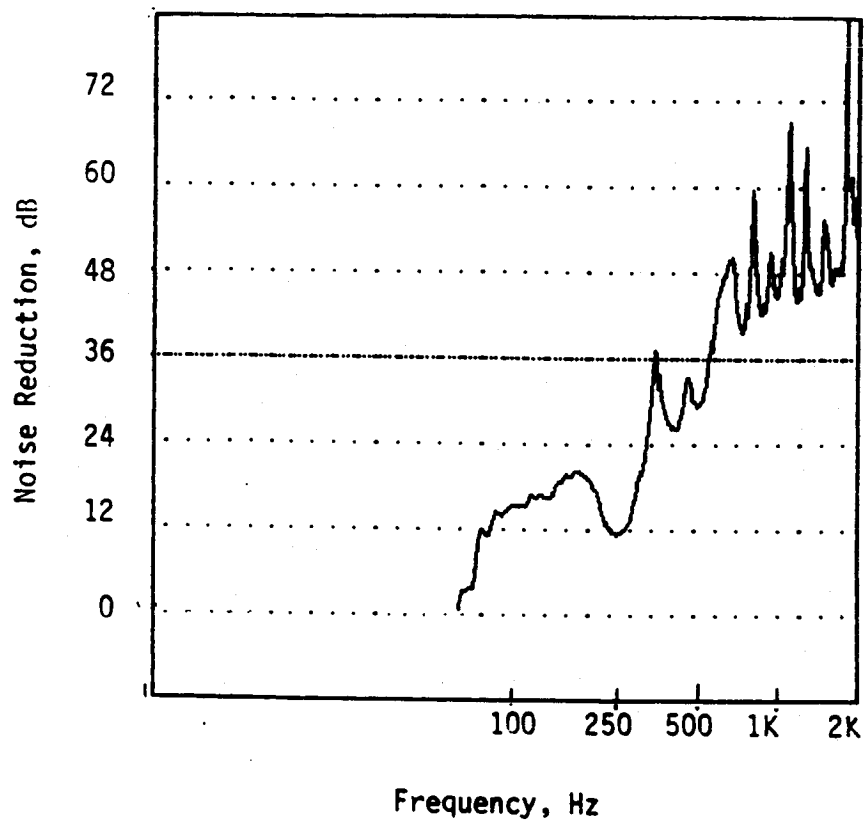


Figure B-12. Noise Reduction of Panel D.2

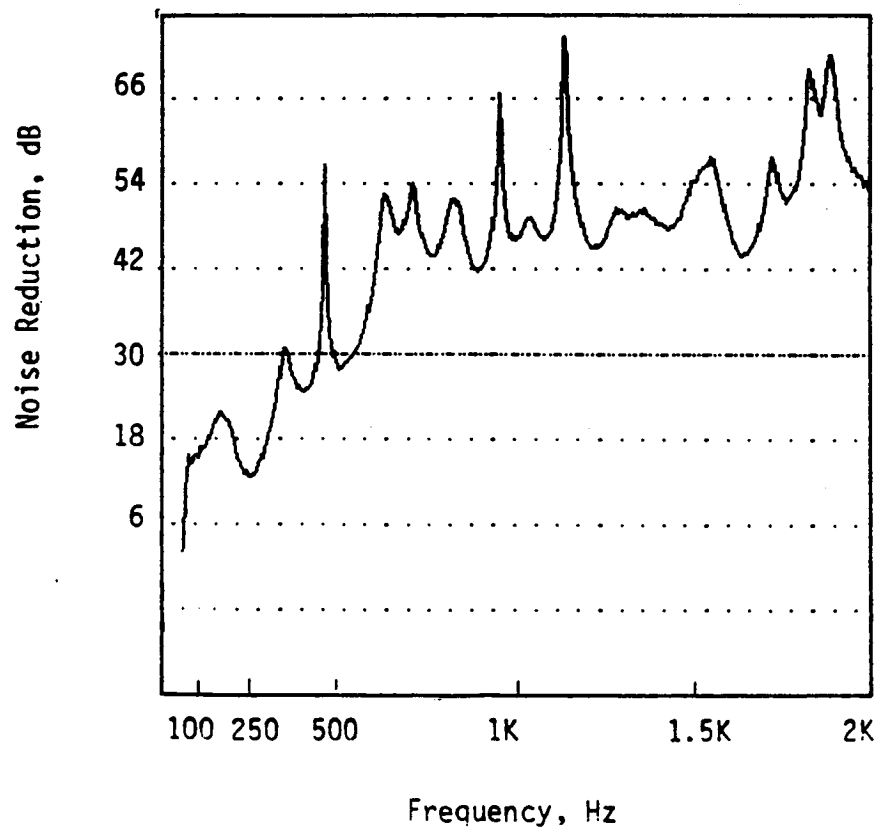
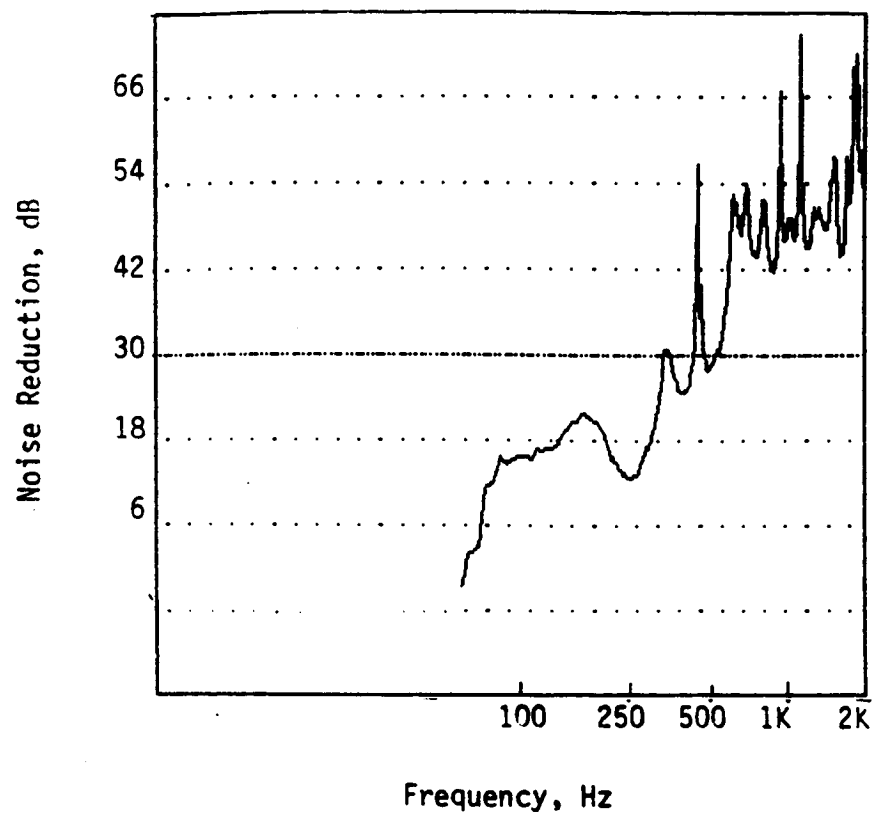


Figure B-13. Noise Reduction of Panel D.3

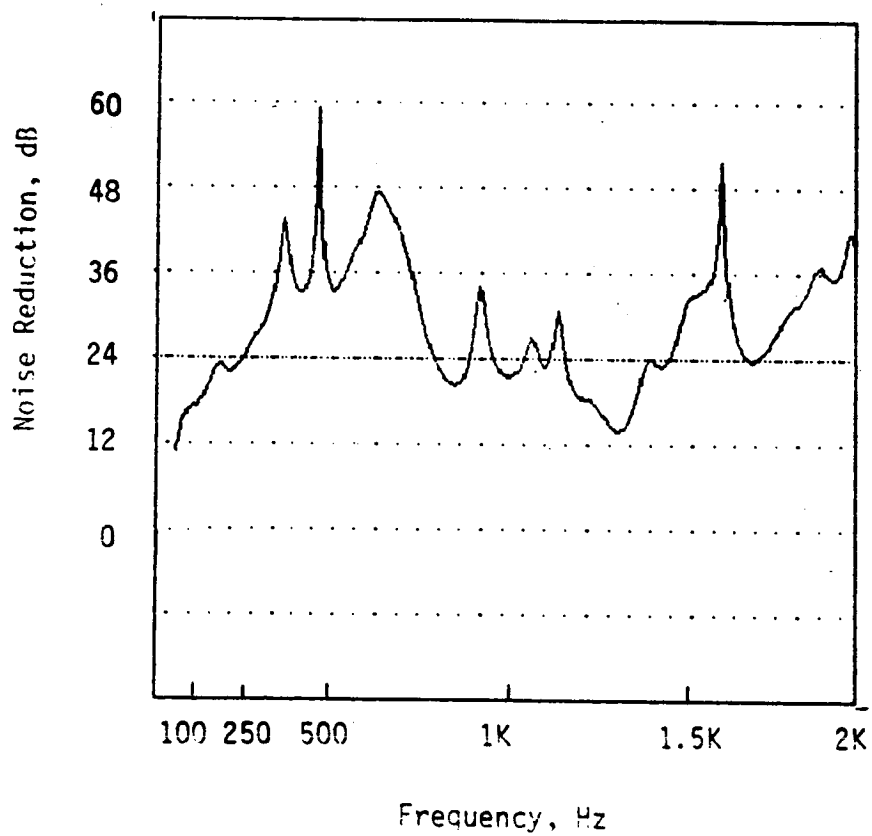
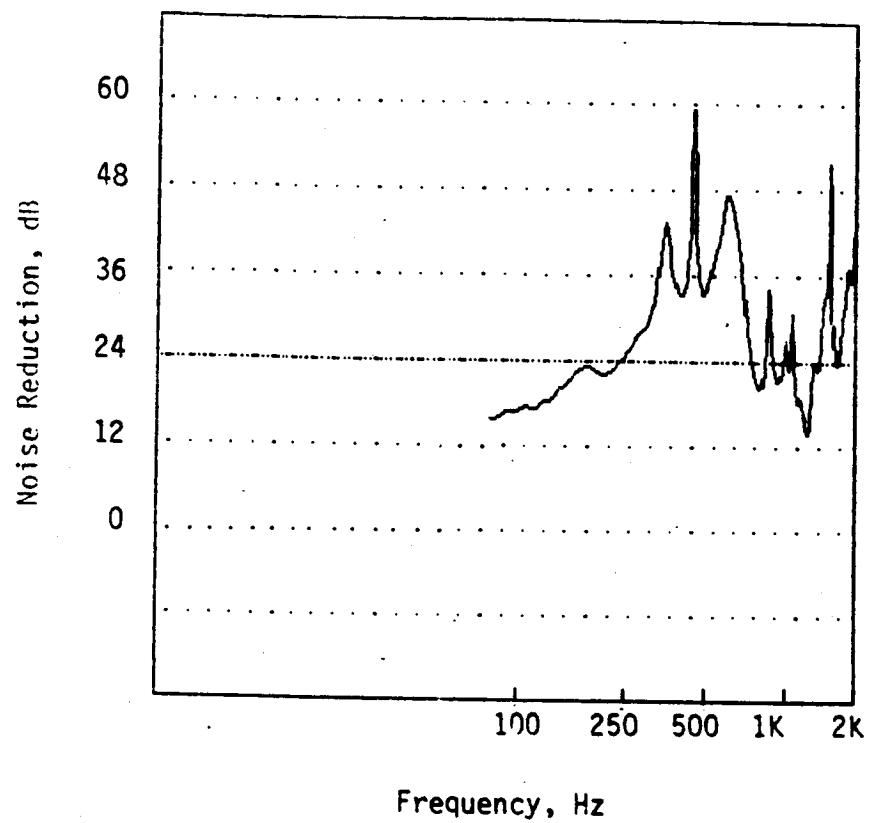


Figure B-14. Noise Reduction of Panel E.2

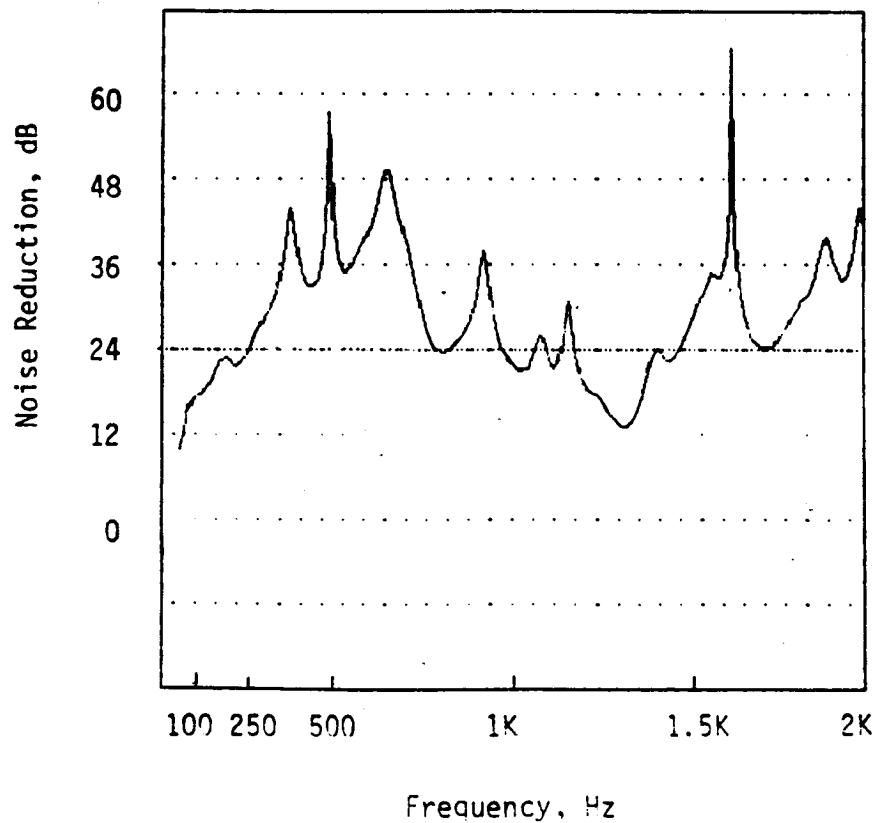
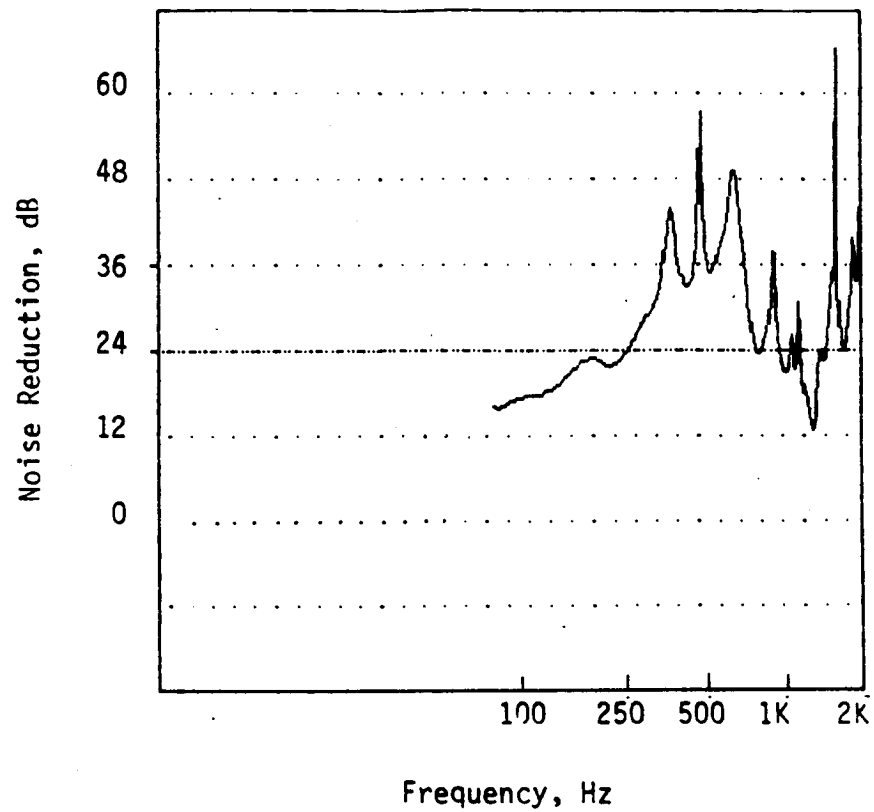


Figure B-15. Noise Reduction of Panel E.3



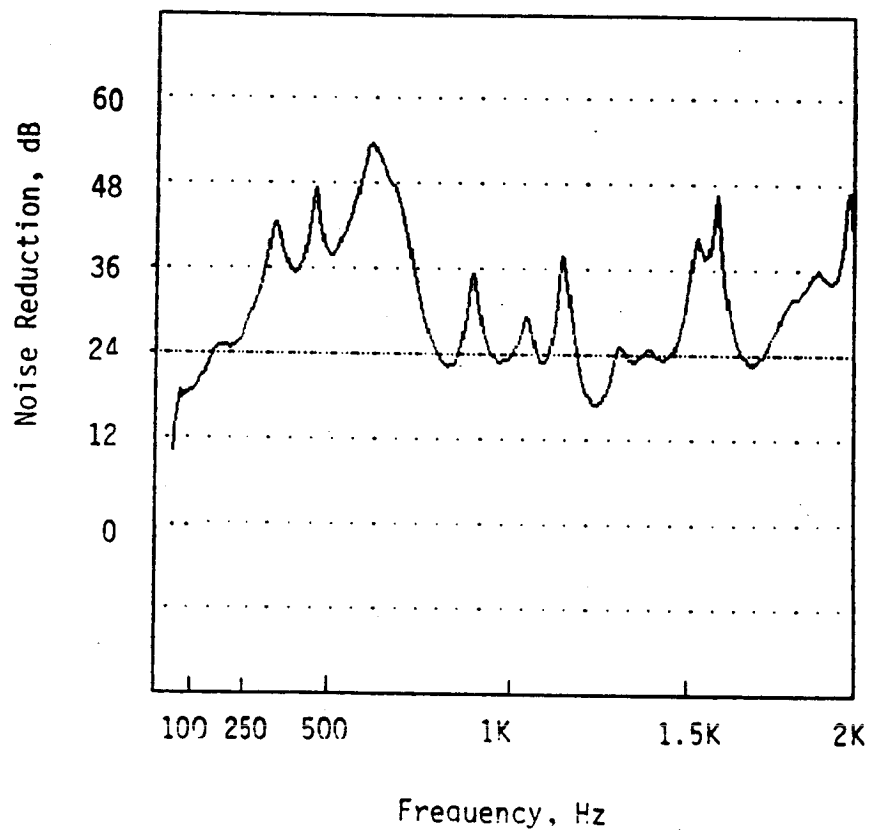
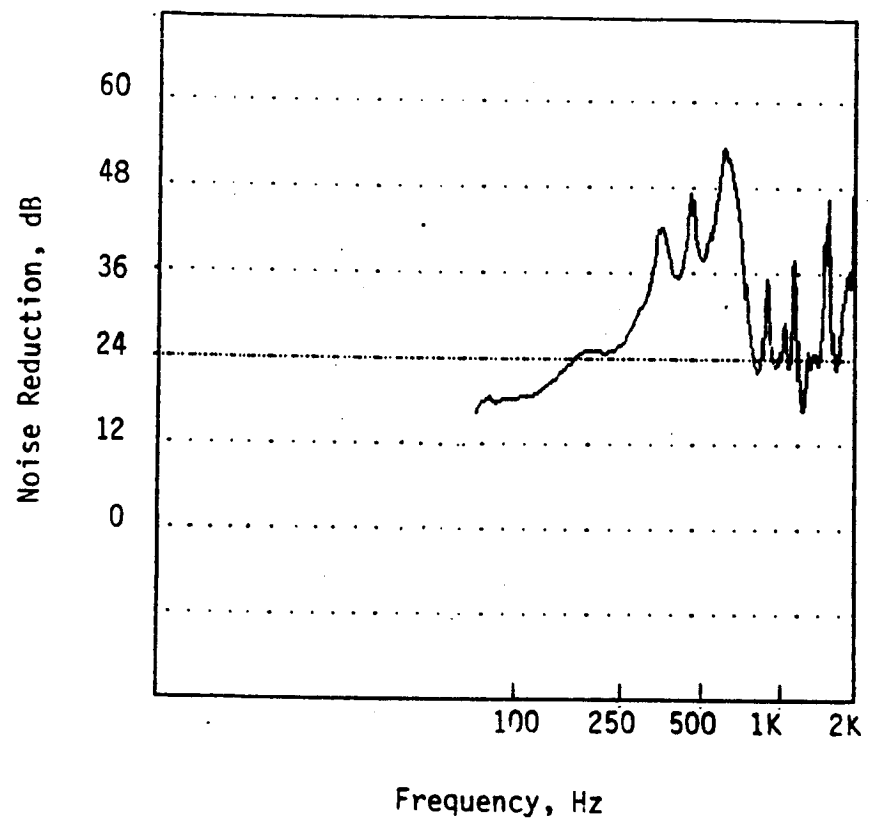


Figure B-16. Noise Reduction of Panel E.4

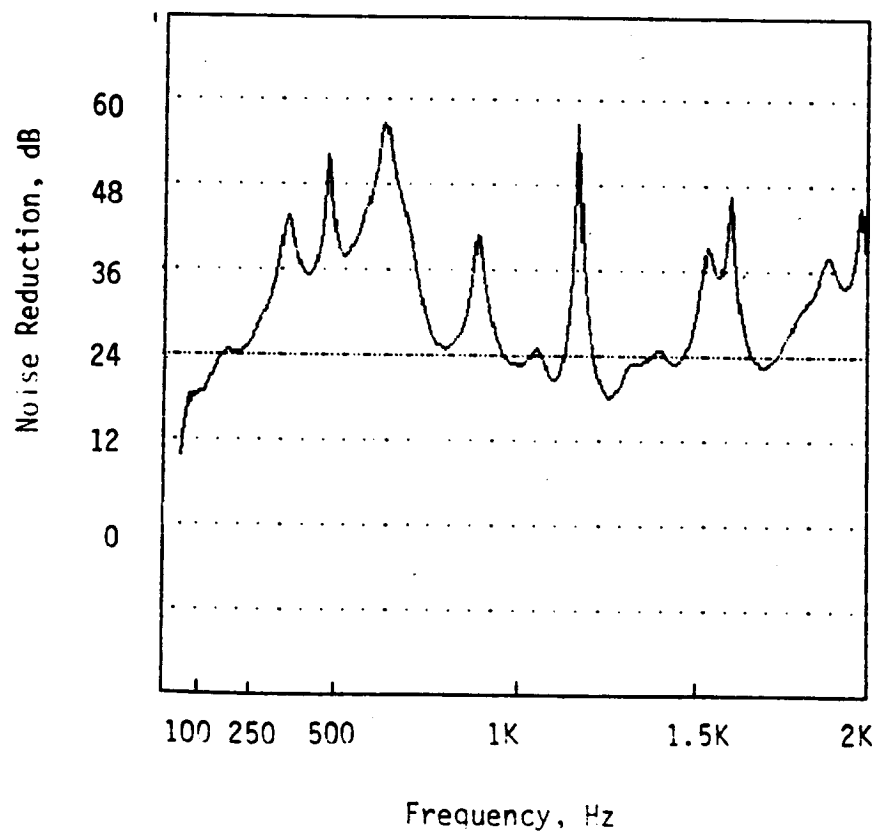
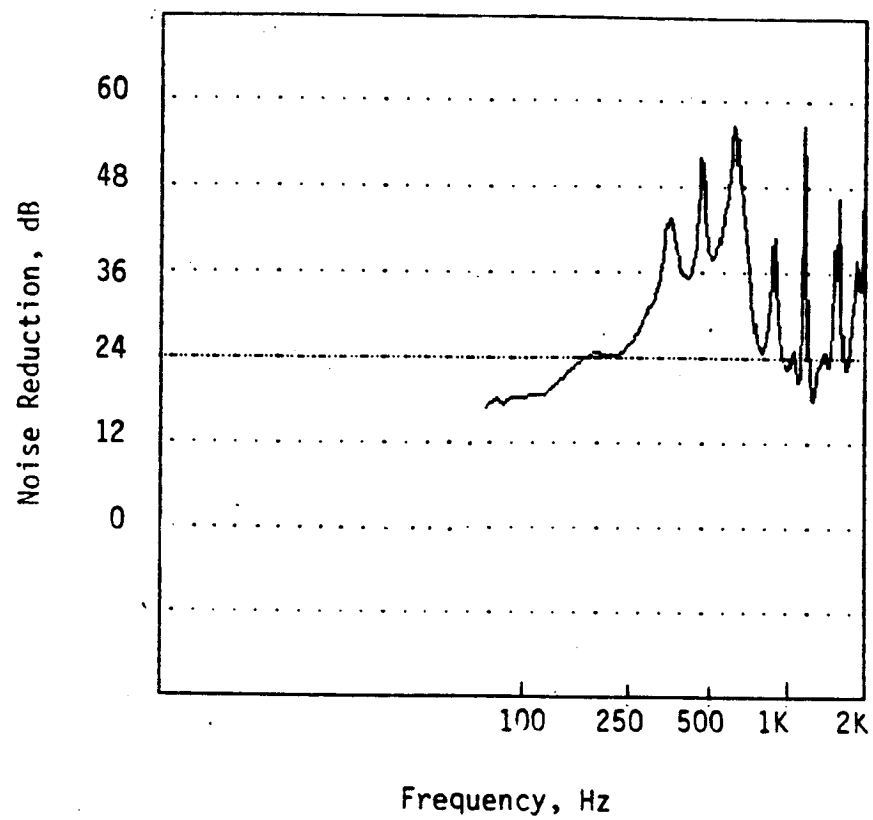


Figure B-17. Noise Reduction of Panel E.5

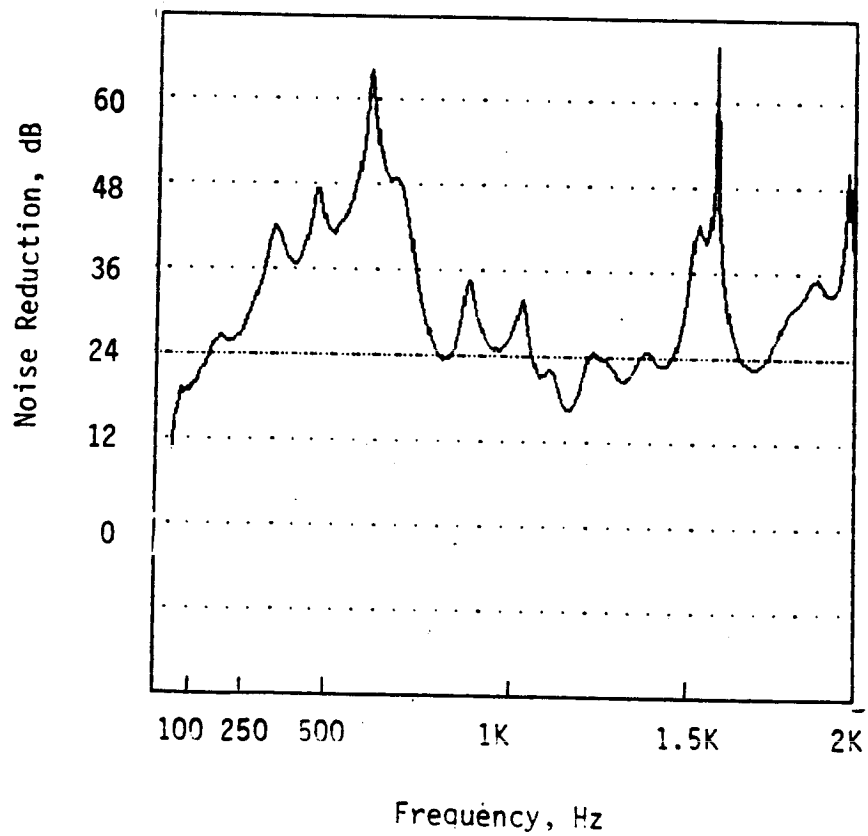
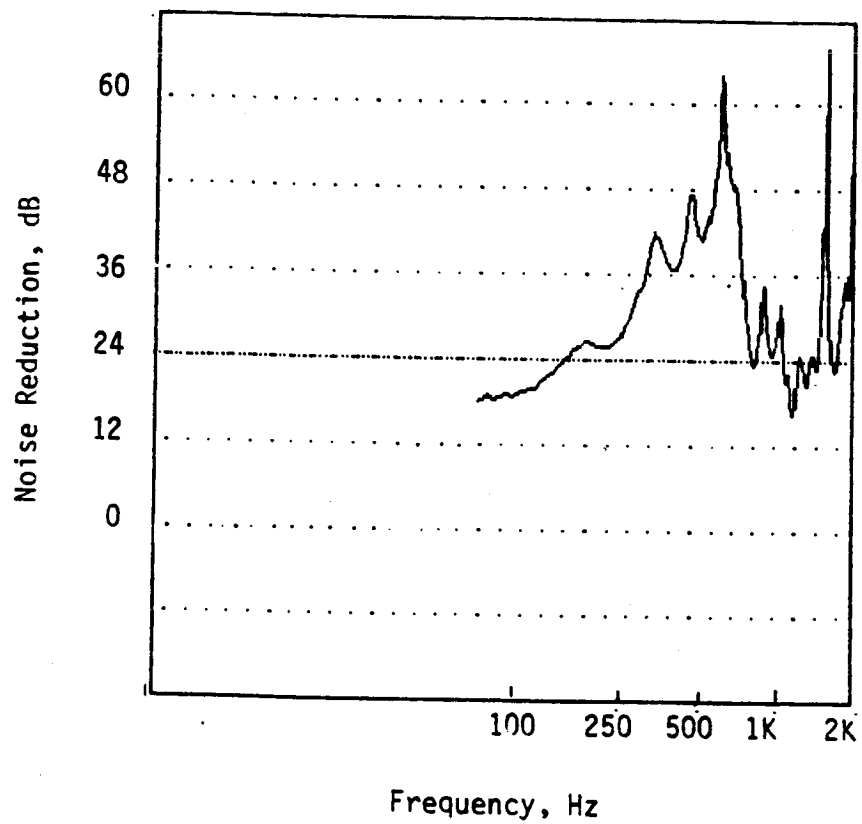


Figure B-18. Noise Reduction of Panel E.6

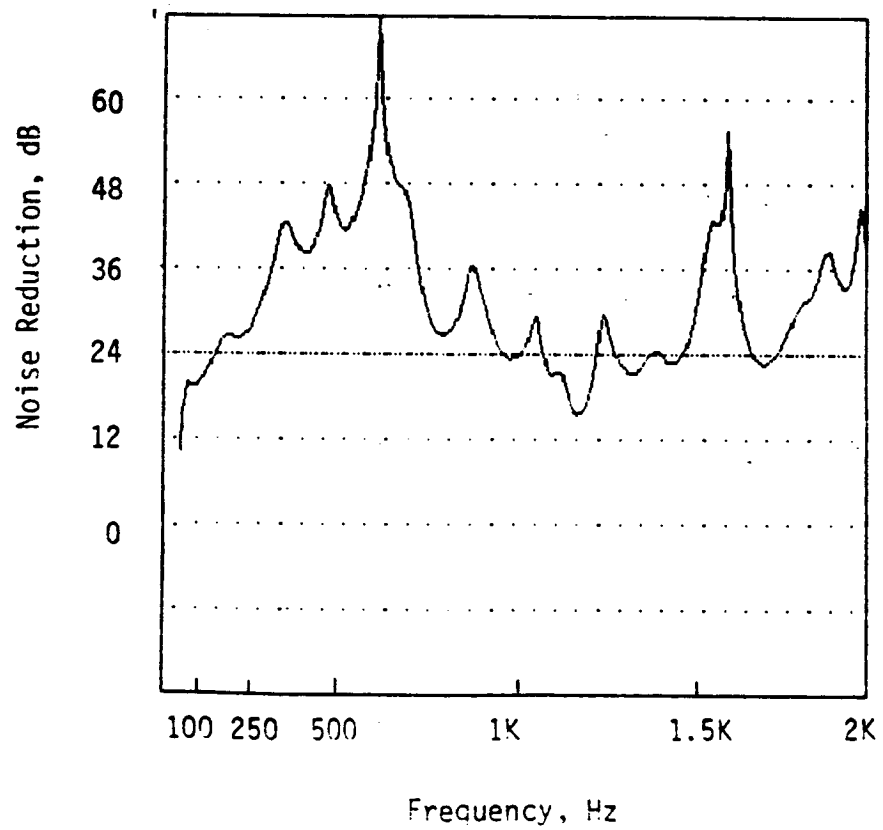
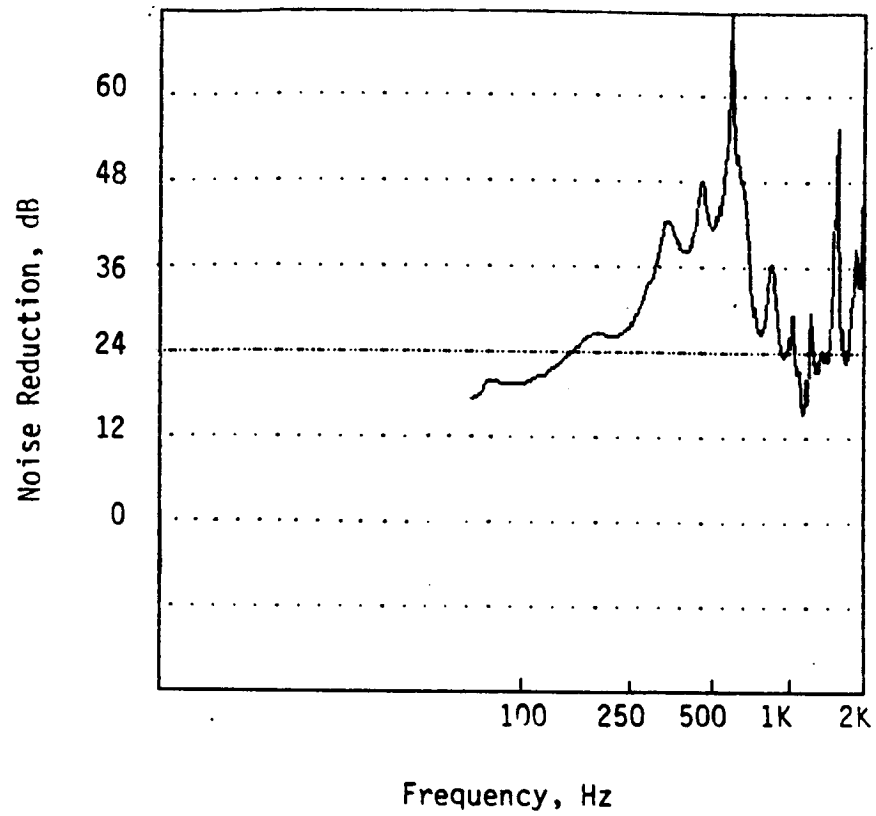


Figure B-19. Noise Reduction of Panel E.7

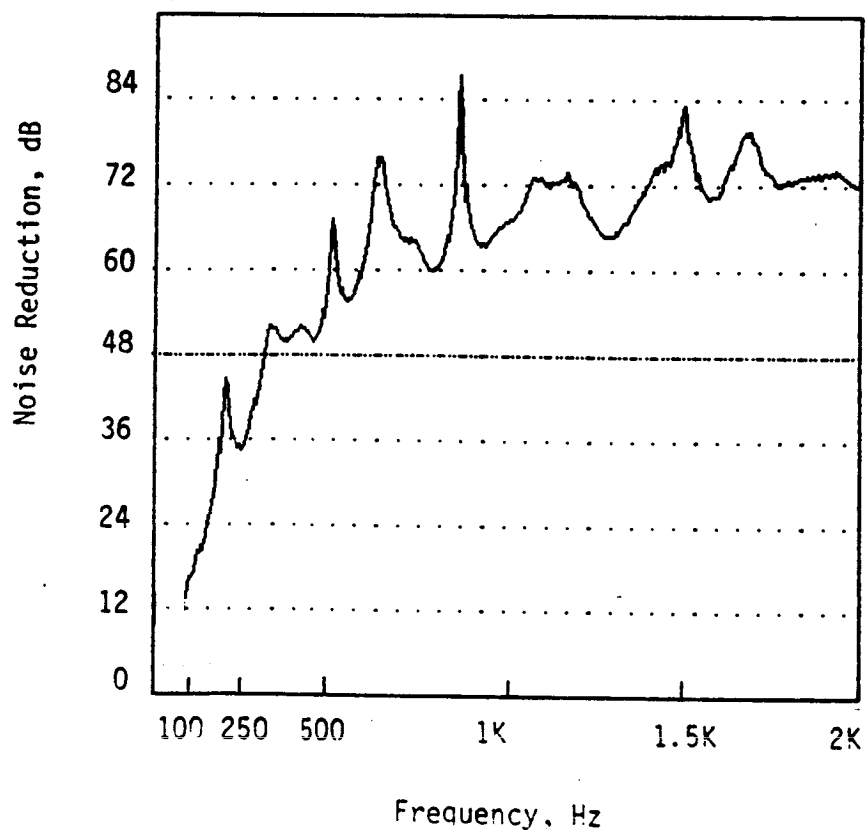
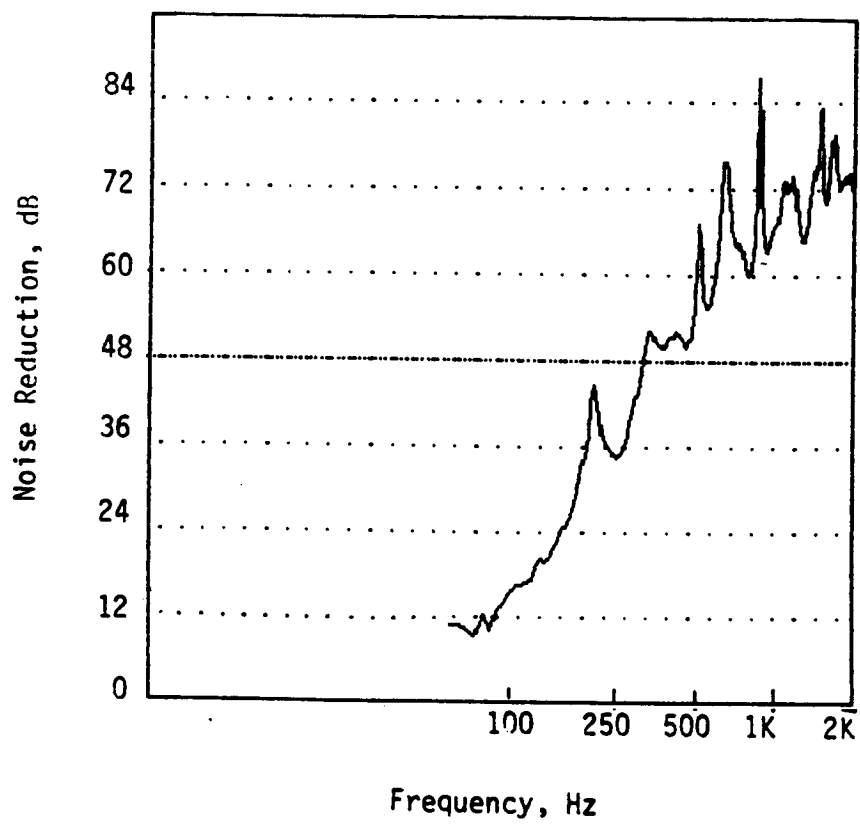


Figure B-20. Noise Reduction of Panel F.1

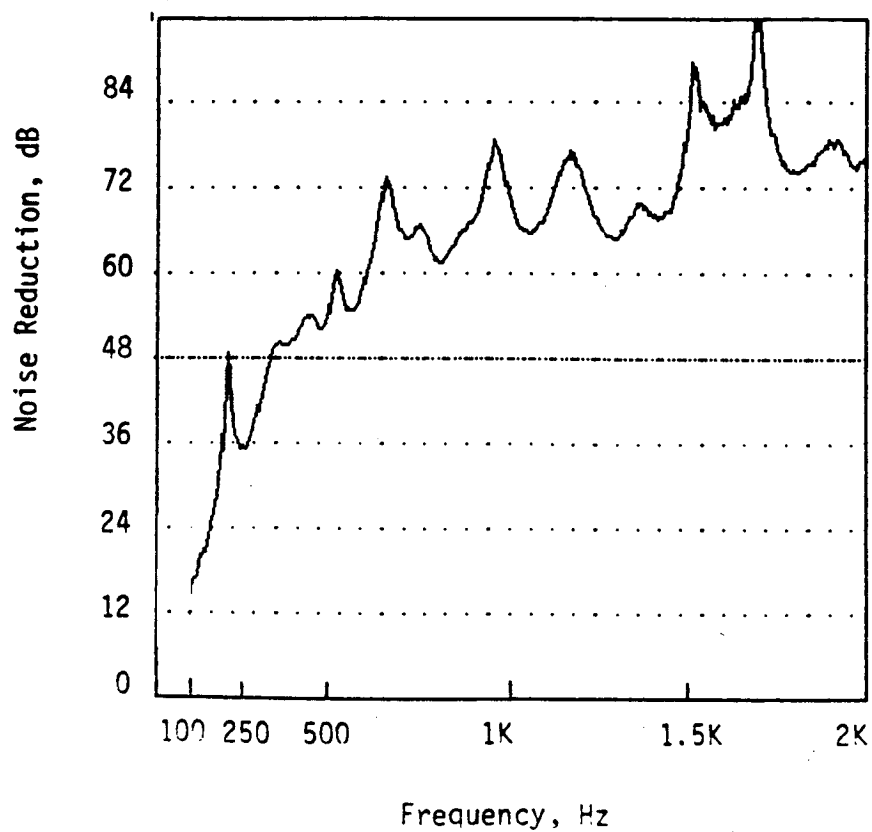
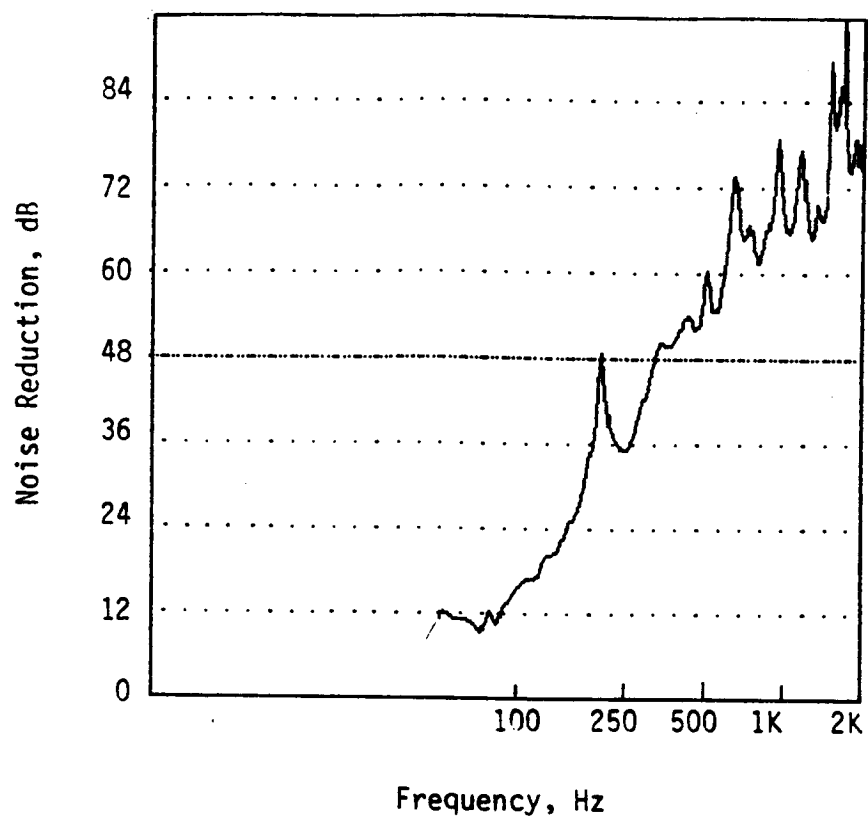


Figure B-21. Noise Reduction of Panel F.2

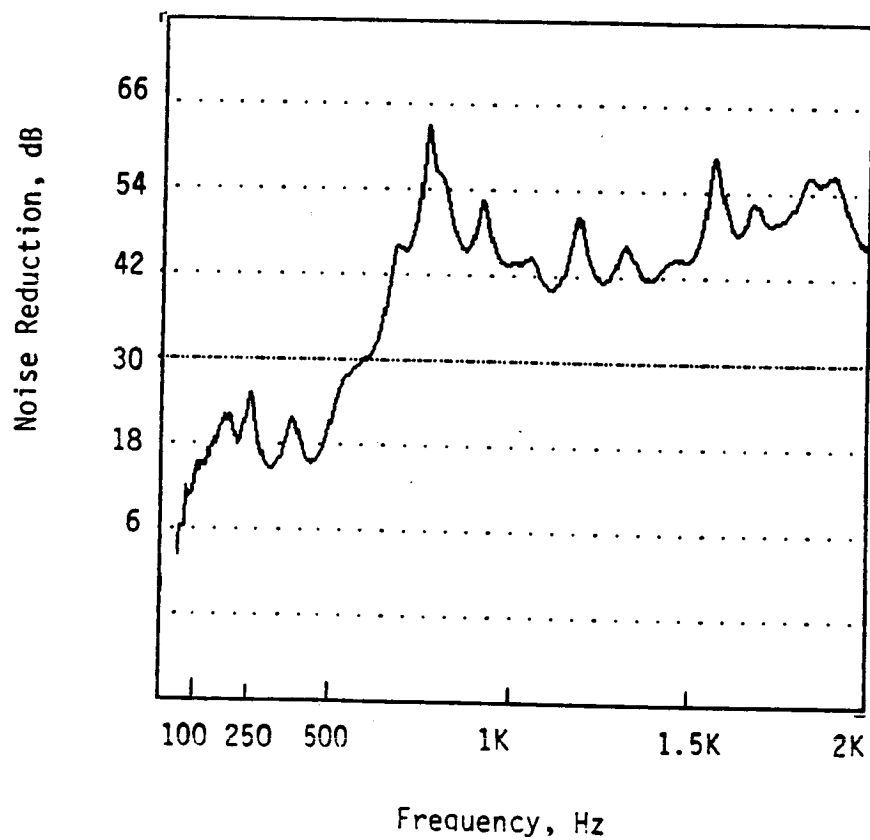
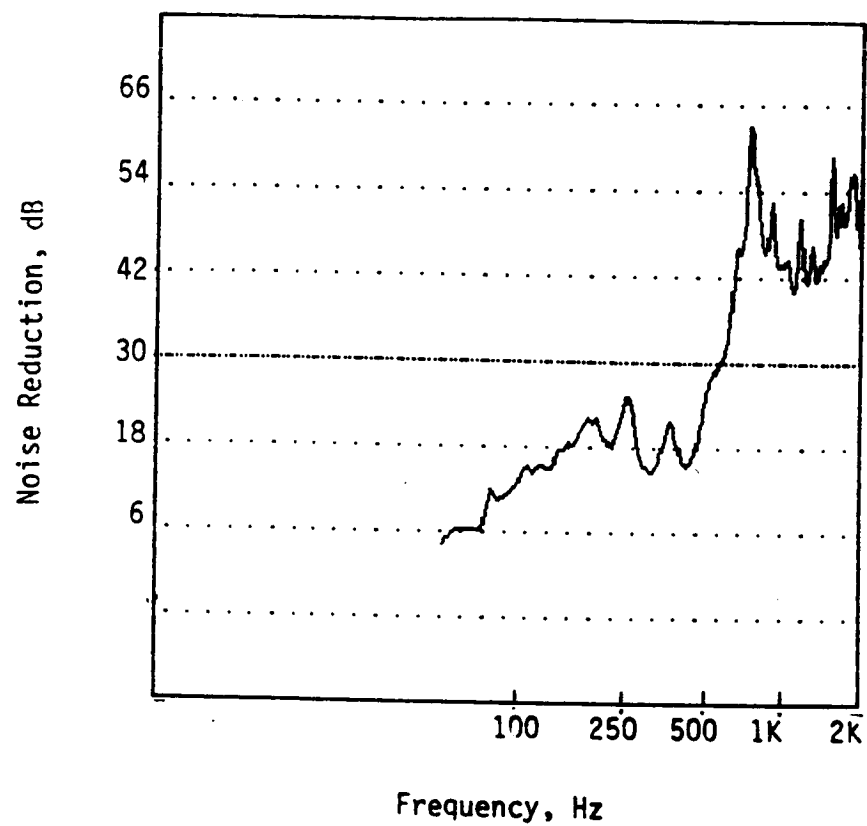


Figure B-22. Noise Reduction of Panel G.1

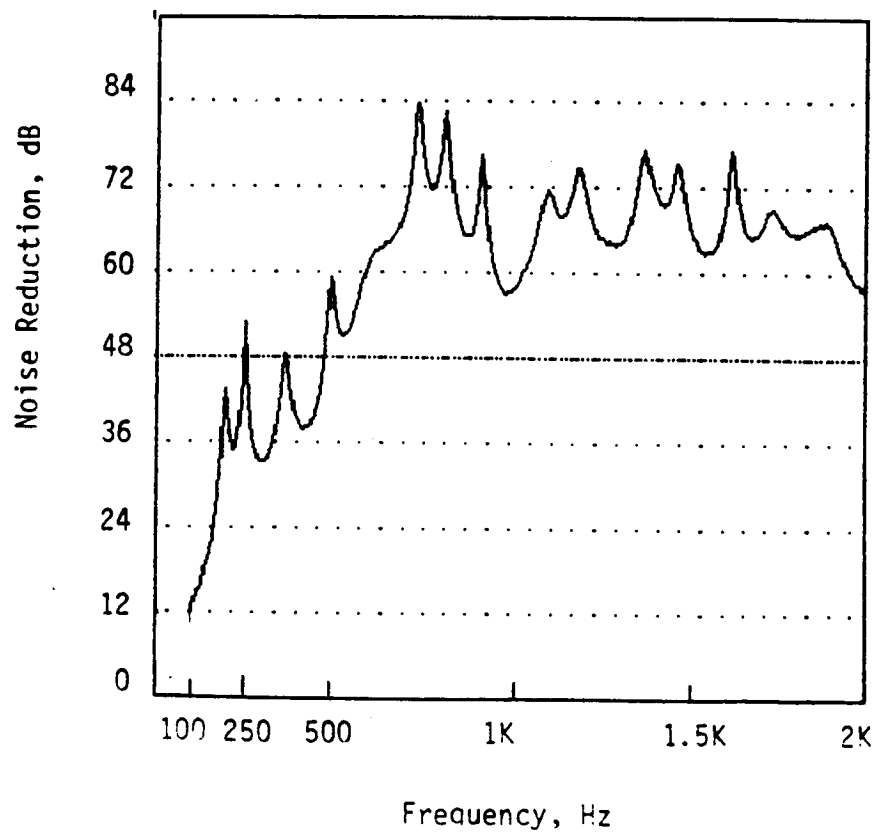
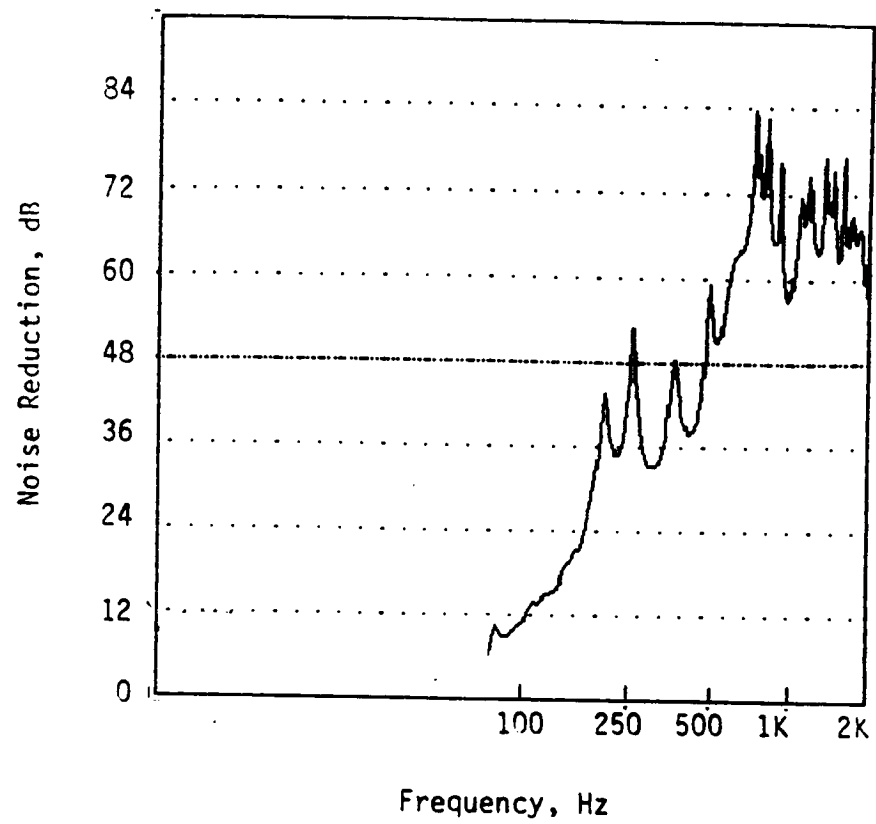


Figure B-23. Noise Reduction of Panel G.2



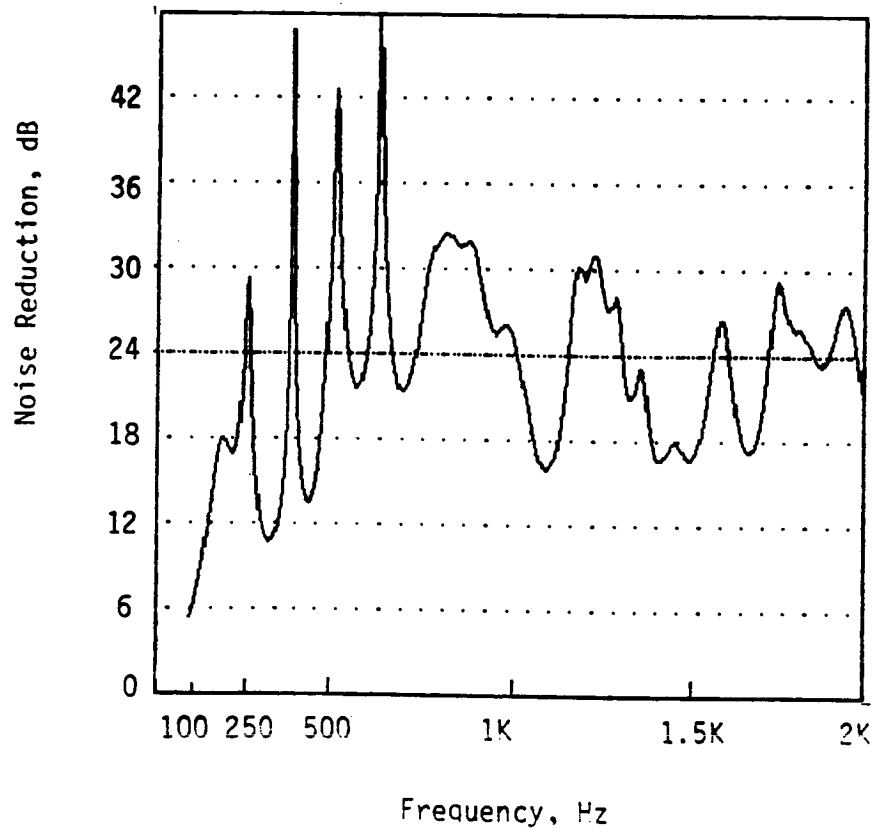
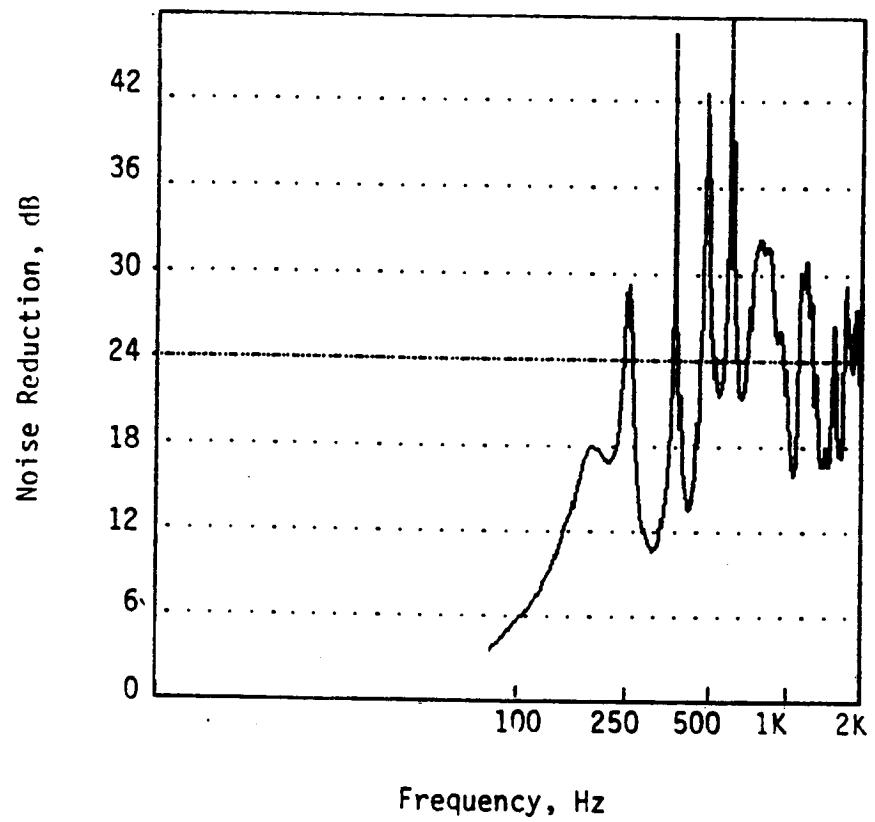


Figure B-24. Noise Reduction of Panel H.1

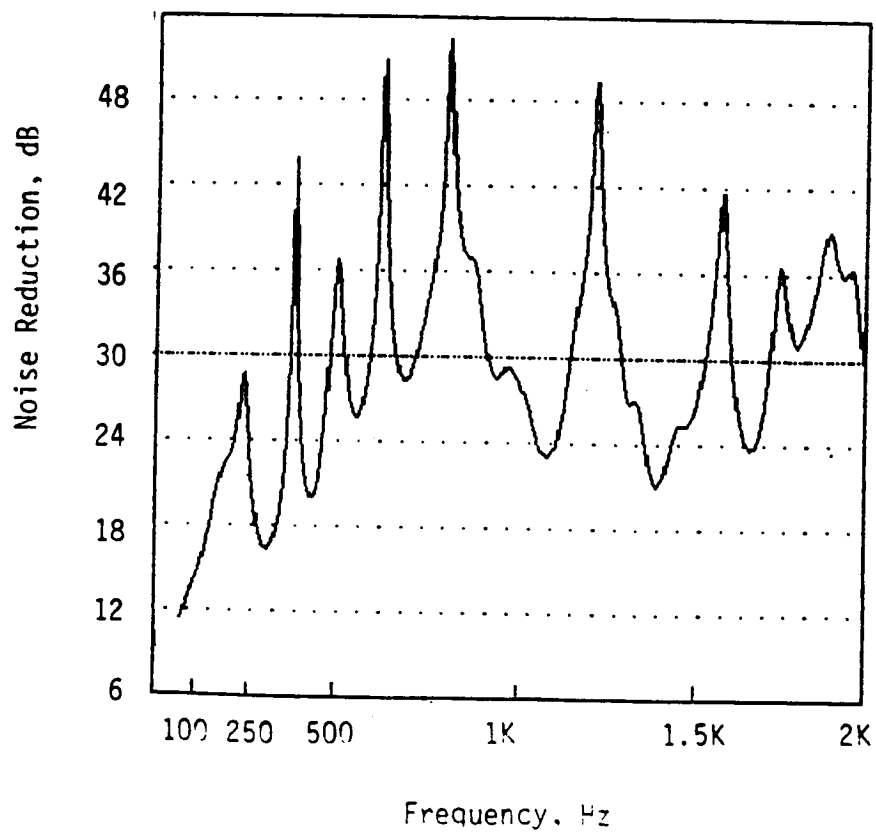
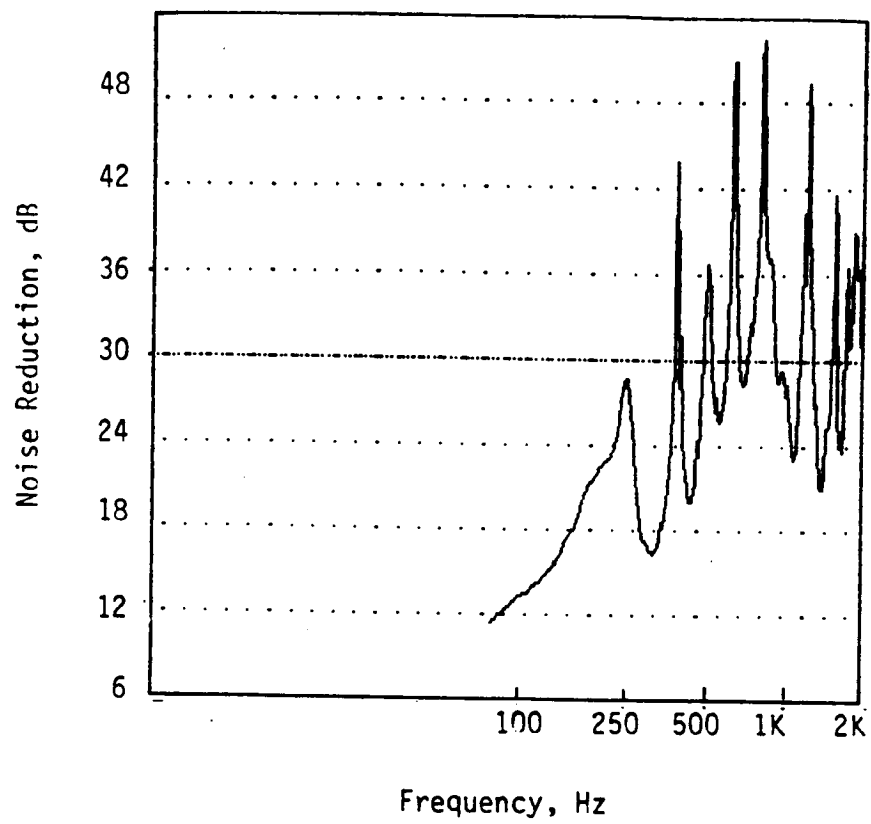


Figure B-25. Noise Reduction of Panel H.2

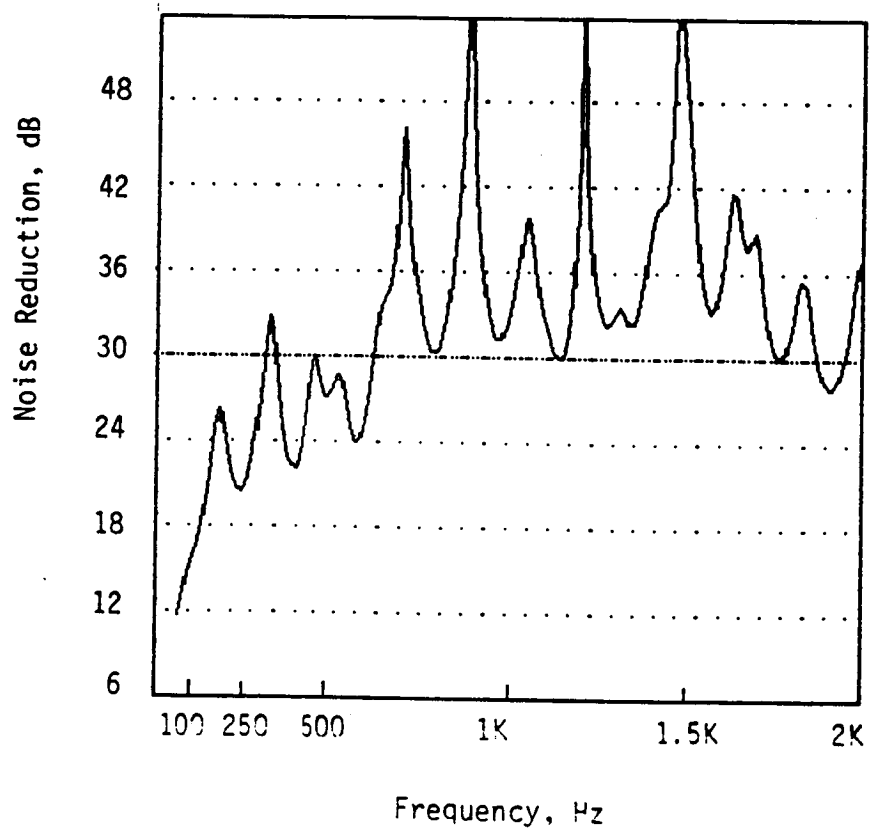
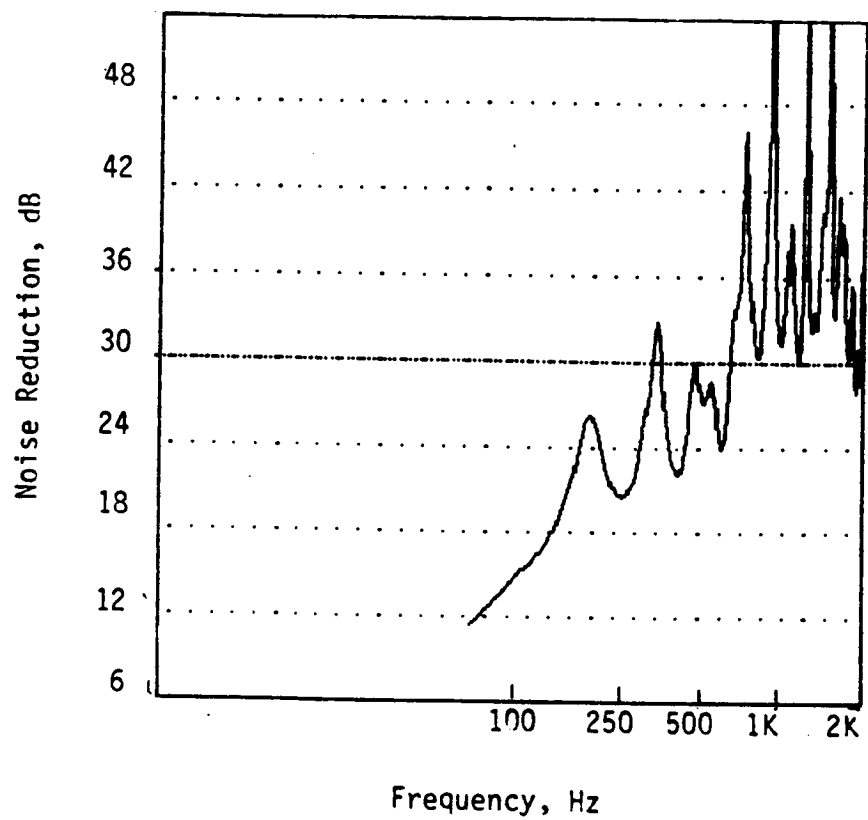


Figure B-26. Noise Reduction of Panel I.1

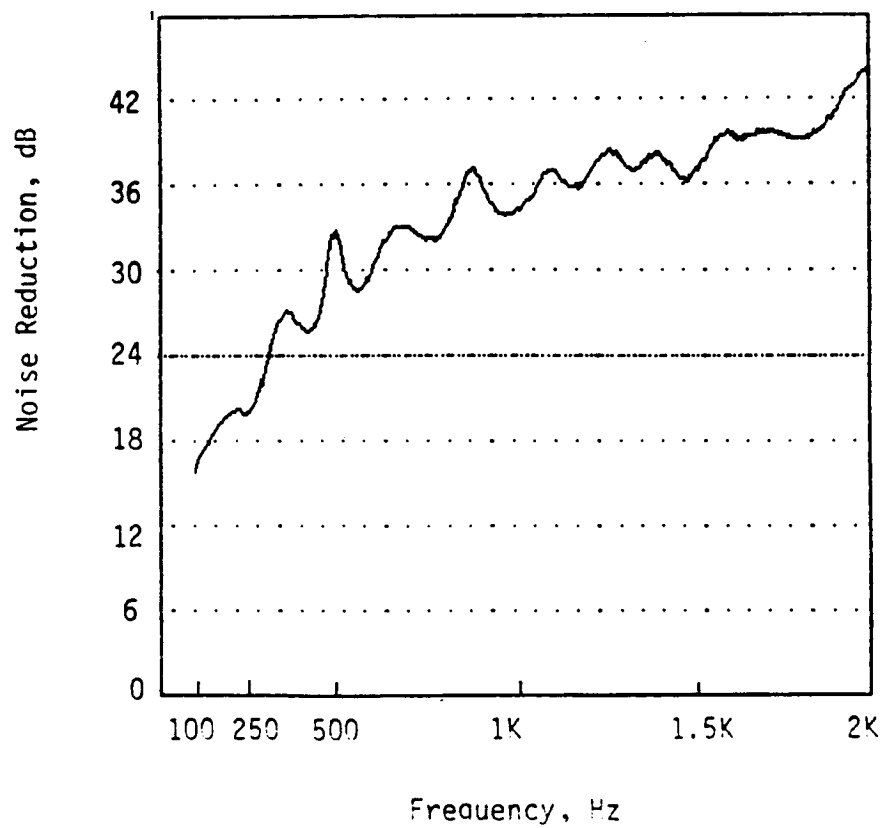
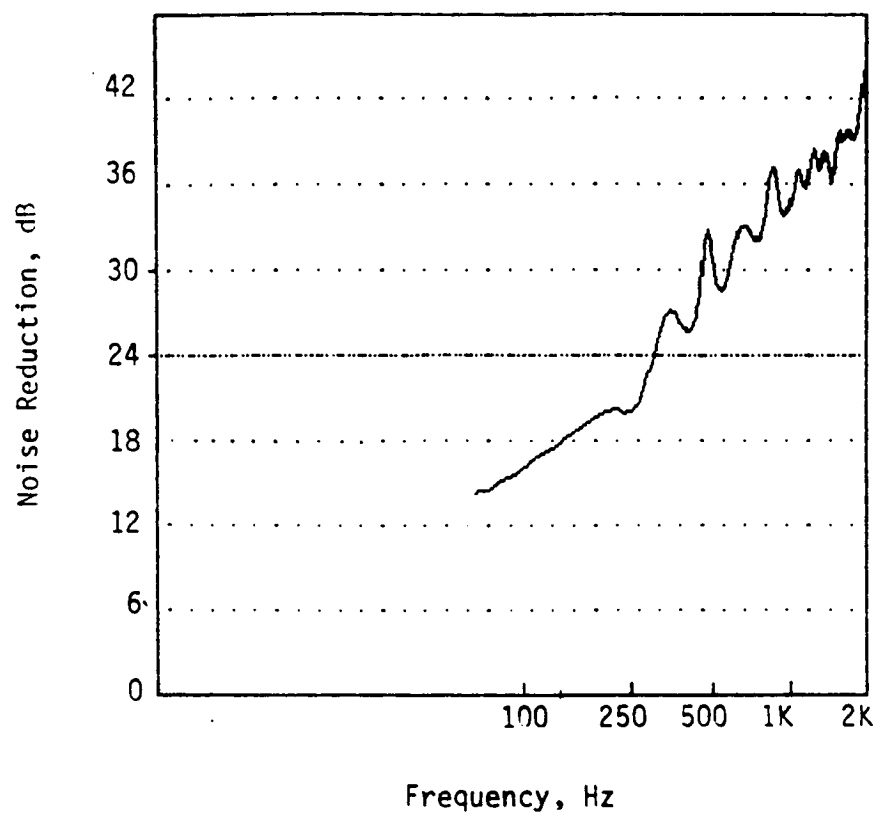


Figure B-27. Noise Reduction of Panel J.1

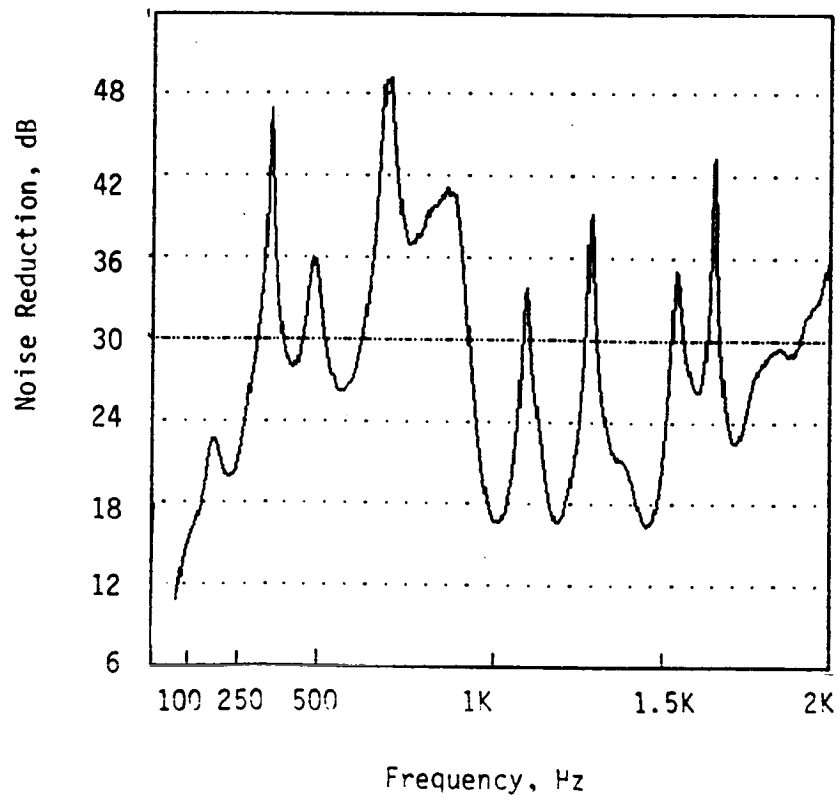
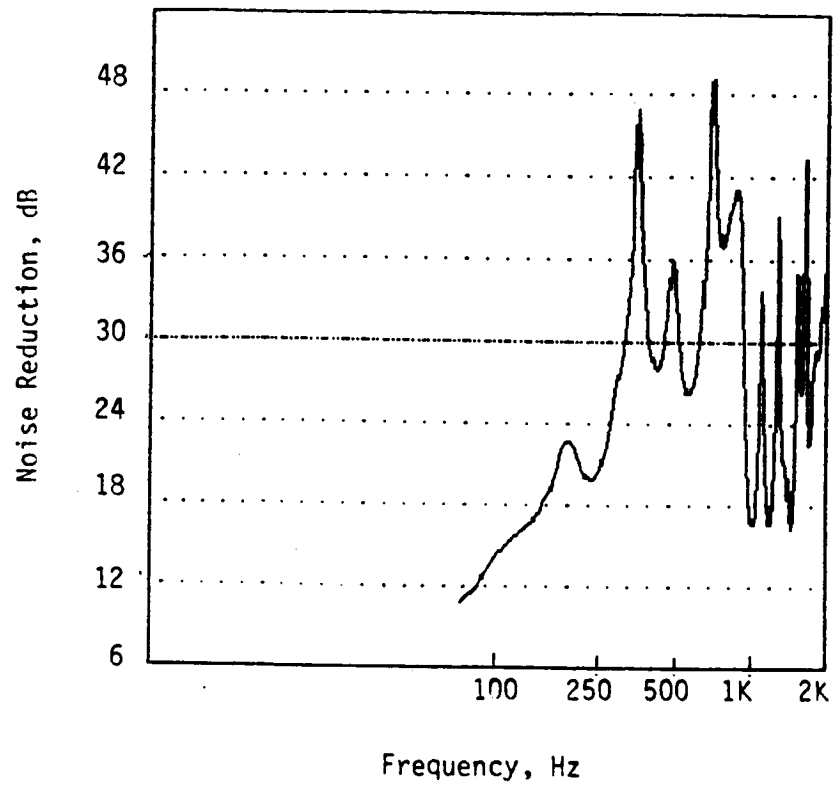



Figure B-28. Noise Reduction of Panel K.1

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16. Abstract  A theoretical and experimental study was performed to devise lightweight sidewalls for future turboprop aircraft. Seven concepts for new sidewalls were analyzed and tested for noise reduction using flat panels of 1.2 m x 1.8 m (4 ft x 6 ft), some of which were aircraft-type constructions and some of which were simpler, easier-to-construct panels to test the functioning of an acoustic principle. Aircraft-application sidewalls were then conceived for each of the seven concepts, and were subjectively evaluated for their ability to meet aircraft nonacoustic design requirements. As a result of the above, the following sidewall concepts were recommended for further investigation: a sidewall in which the interior cavity is vented to ceiling and underfloor areas; sidewalls with wall-mounted resonators, one having a conventional trim panel and one a limp one; and a sidewall with a stiff outer wall and a limp trim panel. These sidewalls appear to promise lower weights than conventional sidewalls adjusted to meet similar acoustic requirements, and further development may prove them to be practical.					
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